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I.

STEEL CRANK SHAFT FORGINGS.

By Assistant Engineer A. M. Hunt, U. S. Navy.

In forging a crank shaft of iron a start is made from a porter bar, on one end of which fagots of iron are piled, heated and welded. This process of accretion is continued, and as the piles of fagots are welded the shaft is roughed into shape.

In the case of a solid forged steel shaft a start is made from a solid ingot, generally cast with a tong hold to which the porter bar is fastened. No welds can be allowed, as they are uncertain elements where steel is used. It is most important that the ingot used should be as solid as possible, free from blow holes and cracks.

The chemical composition of steel used in making large forgings is generally similar to that of steel used for castings in that it contains from .1 to .2 of 1 per cent. of silicon; and as the action of silicon, unless considerable manganese is present, is to render the carbon-iron compounds unstable, with consequent non-uniformity of composition, manganese is added, sometimes as much as 1 per cent. or over. The addition of silicon prevents the occlusion of the gases while the steel is solidifying,

and so gives an ingot free from blow holes. The specifications for the inspection of steel for the new cruisers require that "all steel for forgings must be made by the open hearth process, and must not show more than .06 of 1 per cent. of phosphorus, nor more than .04 of 1 per cent. of sulphur, and must be of good quality in other respects." As a matter of fact most of the manufacturers use steel having considerably less phosphorus than allowed, from a desire to ensure that nothing but the very best forgings may leave their works.

The failure of the Dolphin's steel shaft when that vessel was being tried, and the failure of the Boston's thrust shaft under proof test, put quite a damper on the use of steel shafts in our Navy, but a careful examination of the steel revealed the causes of the breakages, and now thoroughly reliable steel shafts are being made for our ships. Careful inspection insures the quality of the shafts furnished, and emulation between different manufacturers results in their furnishing material which is almost always in advance of contract requirements.

In the case of the broken shafts above mentioned careful analysis showed that the metal at the places where broken was high in phosphorus and other impurities, and tensile tests showed a considerable variation in tensile strength in different parts of the shafts. I have been informed by a person who was employed in the works where these shafts were made, and at the time when they were made, that the melting furnace was too small to give an ingot of sufficient size to allow the rejection of enough of the metal at the upper end to exclude the unsound portion, which is also the portion where the greatest segregation of impurities occurs. In fact the discard was very slight.

At works where a great many heavy forgings are continually being made the ingots are generally bottom cast in iron molds. At many works ingots are frequently cast in molds formed of thin sheet iron surrounded by sand. The first few blows of the hammer dislodge the envelope of sheet iron. A variety of this form of mold which should be severely condemned is in use in at least one establishment where forgings are made for our naval vessels.

Figure 1 illustrates this last-mentioned form of mold. A thin shell of sheet iron, "*c c*," about $\frac{1}{16}$ of an inch thick, is surrounded by a layer of sand from 5 to 6 inches thick, the base of the mold resting on fire brick. On top is placed a cast-iron neck "*A*." An ingot 30 inches in diameter is made with a neck 12 inches in diameter, and the weight of the neck would approximate 8 per cent. of the total weight in the case of an ingot of 26,000 lbs. The metal in "*A*" will solidify very quickly, being in contact with the cast iron, and while the metal in the body of the mold is still molten. The metal in "*A*" having solidified cannot feed the portion immediately below. As a consequence serious piping occurs at "*D*," and unless a very generous discard is made, unsound metal will be worked into the forging. The latest specifications issued by the Steel Inspection Board require a discard of at least 30 per cent. by weight from the top of the ingot in the case of all large and important forgings. The neck for tong holds should be on the lower portion of the ingot as cast, and if it must be at the top, that portion of the mold should be made of sand or brick. The Bethlehem Iron Co. now have their hydraulic press in operation and are using fluid-compressed ingots for many important forgings. When fluid-compressed ingots are used a much smaller discard suffices than when ordinary ingots are used. The ingots used have various sections, some circular, some elliptical, and others polygonal. The same maker will at different times use dissimilar ingots for forgings of the same size and shape, it being apparently not a matter of great importance, so long as the section is large enough to ensure sufficient work in reducing to the forged size.

The following brief description of the forging of a solid, double-throw crank shaft for the gunboat *Petrel* will give an idea of how such a shaft is made. The forging of a single-throw crank shaft or a section of line-shafting is correspondingly simple.

The ingot from which this crank shaft was forged was bottom cast in a sand mold, with tong hold as shown. It weighed, after discard was made, 23,350 lbs. The chemical analysis showed: C., .23; Mn., .57; P., .059, S., .010; Si., .190. Dimensions are shown in Fig. 2.

To the end of the tong hold "*B*" the porter bar was attached. The end of the porter bar was formed into a spoon which embraced two adjacent sides of the square tong hold, to the other two sides of which another spoon was fitted. The edges of the two were bolted firmly together, thus securing the porter bar to the ingot. Arms at right angles to the porter bar were used to turn the ingot around its axis in the bight of an endless chain which ran over a sheave to which was attached the raising and lowering purchase of a heavy jib crane. An ordinary reverberatory heating furnace was used, the fuel being natural gas. The forging was done under a ten-ton steam hammer, with a piston 24 inches in diameter and 7 feet stroke.

The middle portion of the ingot was first heated, and the part marked "*a, b, c, d,*" between the two cranks, cut out. This piece was 14 inches in length and 19 inches in depth, weighing 1,673 lbs. The portion of the shaft "*c, d, e, f,*" between the cranks, was next slightly worked, after which the forward crank was twisted through an angle of 90°. This was accomplished by a succession of light blows on the forward portion of the ingot, the workmen holding against the hammer by means of the arms on the porter bar. As the twisting progressed, the portion of the shaft between the cranks was worked to a cylindrical shape and down to the forged size. It required three heats to accomplish the twisting, which had to be done at a high heat. The method is primitive, and at better equipped works a turning ratchet worked by hydraulic power is employed.

A V-shaped piece, "*m, n, o,*" weighing 240 lbs. was next cut from the ingot, and the forward end of the shaft and forward eccentric worked into shape, the extreme forward end being drawn down to enable the porter bar to be attached. After shifting the porter bar, that part of the ingot forming the afterpart of the shaft was heated, and the tag "*A B,*" weighing 1,760 pounds, cut off. It was then found that the forward crank had sprung back slightly. This was corrected, and a second V-shaped piece, "*x, z, y,*" weighing 212 pounds, cut out, and the after portion of the shaft and the coupling disc formed under the hammer. The weight of the rough-forged shaft was 16,724 pounds,

the blocks cut from the two ends, from which test pieces were taken, weighing 490 and 444 pounds, respectively.

A rough-forged shaft, as it leaves the hammer, has considerable superfluous metal which must be removed by machining. The blocks between the crank webs are so removed, and, as a general thing, contracts for forgings call for them to be rough machined to within $\frac{1}{8}$ inch all around of finished sizes. A careful watch should be kept during this rough turning for defects in the metal.

Two tensile tests were made on $\frac{3}{4}$-inch rounds, rolled from a 4″ x 4″ ingot cast from the same heat as the ingot for the shaft above described, giving results as follows:

E. L. per sq. in.	T. S. per sq. in.	R. of A. per cent.	Elong. per cent. in 8″
46,610	69,490	49.6	24.0
44,850	69,260	47.7	24.1

Tensile test taken from blocks removed from the ends of the forging gave the following results:

		T. S. per sq. in.	R. of A. per cent.	Elong. in 4″ per cent.
After end	At circumference	64,220	25.0	30.1
	At ½ radius	62,160	24.5	30.2
For'd end	At circumference	68,120	21.5	36.7
		70,020	17.0	36.6
	At ½ radius	66,800	24.0	37.5
		66,600	20.0	32.2

A great number of heats were taken in making this forging, owing to its being quite complicated. This, coupled with the fact that it was not annealed, doubtless accounts in a great measure for the variation in the results of the tensile tests.

The following paragraphs relating to the tests of main engine shafting are extracted from the "Specifications for the Inspection

of Steel for use in the Construction of the Hulls and Machinery of Armored *Cruiser No. 2* and *Cruiser* (protected) *No. 6*:"

"89. *Kind of Material.*—All steel for forgings must be made by the open-hearth process, and must not show more than .06 of one per centum of phosphorus, nor more than .04 of one per centum of sulphur, and must be of the best composition in other respects.

"90. *Treatment.*—All forgings must be annealed as a final process unless otherwise directed.

"99. Ingots for main engine shafting will be cast on end, and at least thirty (30) per centum, by weight, will be discarded from the upper end.

"100. Each length of rough-forged shaft must have a piece cut from each end of sufficient size to allow of the removal of specimens for tensile test, parallel with the axis of the shaft.

"101. From the piece removed from the end which was uppermost in the ingot, four test pieces shall be taken—two at the circumference of finished diameter and two at one-half radius from center. Two specimens will be taken at one-half radius from center of the other piece. At the option of the manufacturer, the above pieces may be taken at right angles to the axis and at right angles to the shaft radius. These pieces to be broken under the same conditions as prescribed for tensile tests. In the case of hollow shafting (either forged or bored), the inside specimens will be taken within the finished section prolonged, but as near as practicable to one-half radius from center.

"102. If the couplings are forged on the shaft, test pieces may be taken from a prolongation of the shaft which shall not have received any more reduction than the shaft at its least diameter.

"103. In the case of solid forged crank-shafts, in addition to the test pieces above specified, two test pieces will be cut from the metal slotted out from each crank, one from the surface of the metal slotted out and one at a distance of one-half the radius of the shaft from the plane passing through the axes of the shaft and crank-pin.

"104. In the case of built-up crank-shafts, test pieces will be cut from each separate forging of the shafts.

"105. *Tensile Test.*—Of all the foregoing test pieces, none must show less tensile strength than 58,000 pounds. The average elongation of the four pieces from the upper end must be at least 28 per centum in two (2) inches. The average elongation of the two pieces from the lower end must be at least 28 per centum in two (2) inches. No piece must show less than 24 per centum elongation in two (2) inches.

"106. Should but one piece fall below the minimum requirement in either particular, the Steel Inspection Board may allow two additional specimens to be taken in close proximity to the failing specimen, and the average results of these two specimens shall replace those of the failing specimen, provided neither falls below the minimum requirement in either particular.

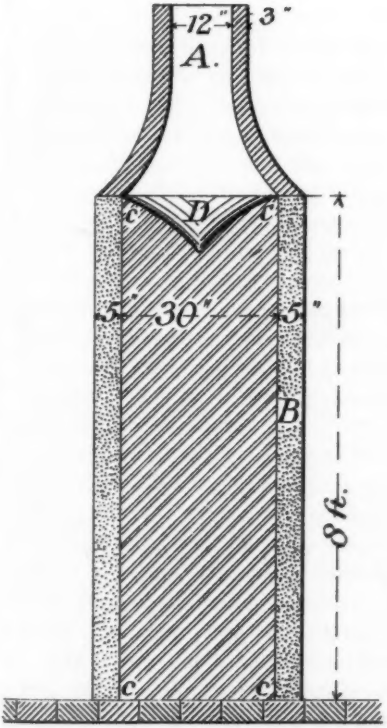
"107. *Quenching Test.*—Bars one-half ($\frac{1}{2}$) inch thick, cut at the outer radius, must stand bending double to an inner diameter of one and one-half ($1\frac{1}{2}$) inches after quenching in water at 82° F., from a dark cherry-red heat, without showing cracks or flaws."

In the case of solid forged crank shafts, the specimens from between the webs have been ordered taken in a plane passing through the axis of the shaft, and at right angles to the plane mentioned in paragraph 103 above. The results of tests taken from several shafts at these points show a very considerably less tensile strength than those taken at the top or bottom. This may be partially due to the fact that the part between the webs does of necessity receive less work than the blocks at the ends from which test pieces are taken, but the segregation of the carbon while the steel is solidifying probably has much more to do with it. When an ingot is poured the metal in contact with the mold solidifies quickly, and this solidification progresses from the sides towards the center, and from the bottom towards the top. As this is going on the elements other than iron rise toward the top. The extreme lower portion of the ingot having solidified before the liquation has proceeded to any extent, is of about average composition; the upper portion of the ingot has

more than the average amount of impurities, and there must be a zone which has less than the average amount of hardening elements. This zone is the central portion of the lower part of the ingot, and in the case of a single throw-crank shaft, forged from the lower block of a big ingot, the pieces cut from between the webs in the positions indicated above would be taken from this zone, and containing less carbon than other portions of the forging, would show less tensile strength. Special attention should be directed to this portion in taking tests. In the case of a crank shaft made from the upper portion of a large ingot, these test pieces should show a higher result than those from the ends, and such was the fact in the only case of which I have a record.

All forgings now inspected under the Steel Inspection Board are required to be annealed. Such treatment is absolutely essential to the production of uniform qualities, and intelligent heat treatment of the steel after the forging is finished is probably as beneficial as the hammer work. By oil tempering and annealing castings can be made to show the physical characteristics of forgings. If it were possible to ensure absolute soundness in castings, forgings would soon be very extensively displaced.

Fig. 1.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A.
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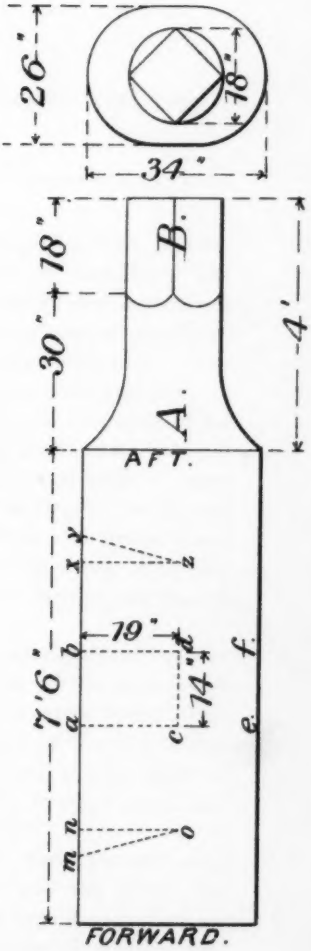

26"
18"
34"
18"
B.
30"
4'
A.
AFT.
x
y
z
19"
b
d
14"
7'6"
a
c
e
f
m
n
o
FORWARD.

Fig. 2.

II.

FITTING UP THE CRANK SHAFTS OF THE U. S. S. *NEWARK.*

By Passed Assistant Engineer Albert C. Engard, U. S. Navy.

The *Newark's* crank shafts are of the built-up type, hollow-forged, journals and crank pins 13 inches in diameter. The shaft, pins, and crank webs are received from the steel mill rough finished to nearly the finished sizes. The crank webs are planed to the finished size and the edges rounded off by hand. They are next laid out for boring the holes for the shaft and pin. A piece of wood is fitted across each hole with a thin strip of tin for a center, and a circle of the diameter of the hole to be bored is scribed and lightly marked at short intervals with a sharp center punch. These circles facilitate the setting of the boring bar. The key ways are laid out and marked in a similar manner, and these having been slotted, the webs are ready for boring. This is done on a horizontal floor boring machine, a pair of webs at a time.

Cast-iron stools or supports have been provided, the bottom of which is flat to set firmly on the bed plate of the boring machine, and the top cast to conform with the crank webs. The webs are placed on the supports, parallel and at the proper distance apart, and after being measured and lined and found to be in the correct position, they are secured by heavy clamps and bolts to the bed plate of the boring machine.

The boring bar is now placed through the holes for the crank pin, set by the circles already described, and the holes are bored to the finished size, the hole in one web being $\frac{1}{16}$ of an inch larger than in the other, for the purpose of slipping the smaller end of the crank pin through easily. The webs are now ready for heating to receive the pin.

The crank pins have a hole through them, in each end of which a plug is fitted for the center on which the pins are turned.

The pin is 13 inches in diameter for the wearing surface, and $13\frac{31}{32}$ inches and $14\frac{1}{32}$ inches in diameter for the ends that go in the crank webs. The ends are turned to within $\frac{1}{8}$ of an inch of the diameter of the holes in the crank webs. The key seats are now laid out, the pin taken to the drill press, and a hole the width and depth of the key way drilled with a square end drill. The inner end of this hole is then squared across to the width and depth of the key seat for the purpose of giving the planing tool clearance. The pin is then put on a planer and the key way finished. The key ways are filled in with hard wood to prevent the pin, when it it is again put in the lathe for turning the ends to the finished size, from jumping by the tool or tearing the edges as the key way passes. The ends of the crank pins are left $\frac{1}{8}$ inch larger than finished size at the first turning, to insure that any roughening of the surface incidental to handling when cutting the key ways may be subsequently removed. The Cramps' allowance for shrinkage is *two papers*, equivalent to about $\frac{1}{125}$ inch.

Two small holes have been drilled in one end and pins inserted, so that a bar can be put across to turn the crank pin and bring key ways fair when in process of shrinking.

All is now ready for shrinking. A perforated cage of light boiler iron is made to fit around the two crank webs for holding the fire for heating them. Charcoal is used for heating, on account of producing little or no smoke. As the webs heat, the holes are carefully measured from time to time in several different places with a steel gauge made the exact size of the ends of the pin, until they have both expanded sufficiently to permit the pin to slip in place easily. The cage and fire are taken away and the holes carefully wiped out to free them from all dirt and dust.

The pin in the meantime has been hoisted in a sling and placed in position for entering the holes. It is pushed in place, the key ways are brought fair, and a temporary key is placed in the key way in one end of the pin. The pin being in its correct position, water is run on the webs to cool them, care being taken to insure both webs being cooled at the same rate, so that they will grip the pin at the same time. After cooling, the webs are bored for the crank

shafts, and the keys fitted and driven in place in the crank pins. The keys are without taper, about two inches longer than required when finished (for driving), and made to fit the key ways neat on top and bottom and a driving fit on the sides. After the keys are driven home the projections used for driving are cut off. The two webs with their crank pin are now set in a vertical position with the flat side of one web resting on a floor plate for the purpose of shrinking on one piece of the crank shaft. The end of the crank shaft has been turned to fit the hole, and the key way planed. A cage to hold the fire, similar to the one used to heat the webs for the crank pin, is placed under the upper web and supported by brick laid on the lower one. The fire is started, and the crank-pin end of the web is kept cool by running water on it. The hole for the shaft is measured from time to time, as mentioned before for the crank pins. When it has expanded sufficiently the cage and the fire are taken away and the hole carefully cleaned. In the meantime the crank shaft has been hoisted with an overhead crane and brought in position for lowering into place. Shores are put between the webs, overlapping the hole, to support the upper web and prevent the shaft from going too far through. A screw jack is also placed under the center of end of the shaft for the purpose of raising it quickly in case it should go too far through the web. A temporary key is put in the key way, and the shaft lowered in place. When found to be in its true position water is run on and the web cooled. The key for this crank web is now fitted and driven (the key is driven with an iron bar through the hole in the other web) before the next piece of shaft is shrunk in place. The key securing the second section of shaft is driven by iron blocking and wedges. A stiff shore is placed between the webs, and heavy clamps and bolts secure them to prevent straining the crank pin while driving the key. The same operation of shrinking and driving the keys is repeated until the shaft is finished. The crank shaft when complete has three crank pins, six webs and four pieces of shaft. The crank pins are set 120 degrees apart. The key ways in the respective shafts and webs having been planed to this angle, it is only necessary to bring them fair to have the pins in their cor-

rect positions. The crank shaft, complete, is now put in a lathe and the journals turned to size. A cast-iron mandrel is made of the size of the crank shaft, and of sufficient length to reach over all the crank shaft bearings. The bearings are fitted to this mandrel, and after being scraped true, the crank shaft journals are fitted to the bearings.

III.

THE CAUSES OF THE VIBRATIONS OF SCREW STEAMERS.

By Assistant Naval Constructor D. W. Taylor, U. S. Navy.

The subject of steamship vibrations, while exceedingly interesting, is somewhat difficult, and there seems to be almost no experimental data recorded on the subject. Without such data a discussion of the general subject would be of little practical use, so I propose at present to discuss the causes producing vibrations and to endeavor to determine measures which will tend to minimize these causes and thus quell the vibrations at their source.

Vibrations are caused almost altogether by forces called into play through the action of the machinery. The action of the sea causes ships to vibrate more or less; but such vibrations are more of the nature of local tremors than oscillations of the structure as a whole.

I shall divide the forces producing vibrations into two classes:

1. Those acting in the engine room and transmitted through the engine framing to the structure of the ship.

2. Those acting upon the propeller and transmitted through the bearings of the shaft to the structure of the ship.

The forces appearing in the engine room are due to the inertia of the reciprocating parts.

The pressures of the steam upon the piston and upon the cylinder heads are practically equal and opposite. If there were no motion taking place these pressures would neutralize each other through the engine framing, and there would be no external force which would produce any strain upon the engine bearers. Owing, however, to the fact that the pressure of the steam upon the piston is virtually increased or decreased at any instant by the force which is necessary to produce the decrease or increase of velocity which the reciprocating masses are undergoing at that instant, there is an unbalanced force called into play which is commonly called the force due to inertia, or the inertia force.

Consider first the case of a single cylinder for which the reciprocating weights weigh W pounds.

The reciprocating weight is reasonably taken as that of the piston, piston-rod, crosshead and one-half the connecting rod, the other half of the connecting rod being taken as rotating uniformly.

Let θ denote the angle made by the crank arm with the axis of the cylinder measured from the extreme throw of the crank from the cylinder. Let C denote the length of the crank arm. Then neglecting the obliquity of the connecting rod, $C \cos \theta$ will denote the distance of the reciprocating weight from its mean position.

Let ω denote the uniform angular velocity of the crank arm. Then $-C \cos \theta \, \omega^2$ is the acceleration of the reciprocating weight.

Denote now by P the unbalanced force equal and opposite to that producing the acceleration. It is convenient to take P as positive when acting in the direction from the cylinder toward the crank, *i. e.*, when acting down for a vertical engine. Then by a well known theorem of dynamics $P = -\frac{W}{g}(-C \cos \theta \, \omega^2)$ $= \frac{WC \cos \omega^2}{g}$. If g, the acceleration due to gravity, is expressed in feet per second, we must express C in feet, and ω, the angular

velocity, must be expressed as the units of circular measure swept through per second. Now one revolution per minute corresponds to an angular velocity per second of $\frac{2\pi}{60} = .10472$.

Then if r denote revolutions per minute,

$$\omega = .10472\, r.$$
$$\omega^2 = .010966\, r^2.$$

Then taking $g = 32.16$ we have with sufficient approximation:

$$P = + .000341\ C r^2 W \cos\theta.$$

Suppose, now, we have n cylinders driving the same shaft, their axes being distant $l_1, l_2 \ldots l_n$ feet respectively from a fixed point O on the shaft. Suppose their cranks are set at angles $a_1, a_2 \ldots a_n$, with a fixed radius of the shaft. Let $w_1, w_2 \ldots w_n$ denote the reciprocating weights, c the common crank arm, and r the common revolutions per minute. Let the angle made by the fixed radius of the shaft (from which $a_1, a_2 \ldots a_n$ are measured) with the plane through the axes of the cylinders be denoted by θ.

Then the system of unbalanced forces producing the accelerations of the various reciprocating weights may be reduced to a single force at O and a couple, by placing at O pairs of equal and opposite forces parallel to each unbalanced force acting.

If P denote the resultant force through O and M the resultant couple

$$P = + .000341\ cr^2 [w_1 \cos(\theta + a_1) + w_2 \cos(\theta + a_2) + \ldots + w_n \cos(\theta + a_n)].$$
$$M = + .000341\ cr^2 [w_1 l_1 \cos(\theta + a_1) + w_2 l_2 \cos(\theta + a_2) + \ldots + w_n l_n \cos(\theta + a_n)].$$

Expanding and rewriting,

$$P = + .000341\ cr^2 [(w_1 \cos a_1 + w_2 \cos a_2 + \ldots + w_n \cos a_n)\cos\theta - (w_1 \sin a_1 + w_2 \sin a_2 + \ldots + w_n \sin a_n)\sin\theta].$$
$$M = + .000341\ cr^2 [(w_1 l_1 \cos a_1 + w_2 l_2 \cos a_2 + \ldots + w_n l_n \cos a_n)\cos\theta - (w_1 l_1 \sin a_1 + w_2 l_2 \sin a_2 + \ldots + w_n l_n \sin a_n)\sin\theta].$$

Now the force P and couple M (whose axis is perpendicular to P) could be reduced to a single force acting at a distance $\frac{M}{P}$ from the origin O. This would be a purely imaginary point of application, and the actual forces would have to be transmitted to it—supposing it on the ship—by the structure of the ship. So, for practical purposes, it is better to suppose P and M both acting, P being always through the origin O.

A convenient position for O is the intersection of the axis of one of the end cylinders with the shaft axis. Then l_1 is always zero, while l_2, l_3 . . l_n have the same sign, being the distance of the second, third, &c., cylinders from the first.

Returning now to the expression for P and M, let us write,

$$\begin{aligned}
w_1 \cos a_1 &+ w_2 \cos a_2 + \ldots w_n \cos a_n = A. \\
w_1 \sin a_1 &+ w_2 \sin a_2 + \ldots w_n \sin a_n = B. \\
w_1 l_1 \cos a_1 &+ w_2 l_2 \cos a_2 + \ldots w_n l_n \cos a_n = C. \\
w_1 l_1 \sin a_1 &+ w_2 l_2 \sin a_2 + \ldots w_n l_n \sin a_n = D.
\end{aligned}$$

In a given case A, B, C and D are readily calculated, but the following graphic method is more instructive:

Take the fixed radius of the shaft from which a_1, a_2, &c., are measured, to be the radius of the first crank. Then a_2, a_3, &c., are the angular intervals between the various cranks and the first one.

Referring to Fig. 1, draw a horizontal line, OP_1, equal on a suitable scale to w_1. Draw $P_1P_2 = w_2$, and inclined at an angle a_2 to the horizontal. Draw $P_2P_3 = w_3$, and inclined at a_3. Continue this process until we have a figure, $OP_1P_2 \ldots P_n$, with as many sides as there are cranks. Fig. 1 is drawn for six cranks. Project P_n upon OP_1 at E. Then evidently $OE = A$; $P_nE = B$.

Similarly for the determination of C and D we have a figure, $OM_1M_2 \ldots M_n$, with sides parallel to the first, but representing on a suitable scale w_1l_1, w_2l_2, w_3l_3, and so on; projecting M_n on OP_1 at G; $C = OG$, $D = M_nG$.

As I have chosen the origin so that $l_1 = 0$, the side $OM_1 = 0$ or M_1 coincides with O, as shown in the figure. But, though OM_1 vanishes, its direction is still that of OP_1; hence M_n is projected on OP_1.

Now let $P_nOP_1 = \beta$, $M_nOP_1 = \gamma$, $OP_n = w$, $OM_n = wl$;
Then

$$A = w \cos \beta,$$
$$B = w \sin \beta,$$
$$C = wl \cos \gamma,$$
$$D = wl \sin \gamma,$$
$$P = + .000341\ cr^2 [w \cos \beta \cos \theta - w \sin \beta \sin \theta].$$
$$M = + .000341\ cr^2 [wl \cos \gamma \cos \theta - wl \sin \gamma \sin \theta].$$

or

$$P = - .000341\ cr^2 w \cos (\theta - \beta).$$
$$M = - .000341\ cr^2 wl \cos (\theta - \gamma).$$

The figure $OP_1P_2 \ldots P_n$ may well be called the Force Polygon, and $OM_1M_2 \ldots M_n$ the Moment Polygon.

I wish now to call attention to Figs. 2, 3, 4 and 5, which are the Force and Moment Polygons of one engine of the U. S. vessels *Charleston*, *Yorktown*, *Vesuvius* and *Cushing* (torpedo boat). These vessels are all twin screw. The necessary particulars are found in Table 1. The reciprocating weights given are obtained from the total weights of reciprocating parts for each cylinder by subtracting about one-half the weight of connecting rod. The revolutions are about the maximum maintained on trial. It should be noted that the figures are drawn to different scales.

The results thus obtained are very instructive, and show clearly the advantage as regards inertia strains of increasing the number of cylinders.

It may be noted that the resultant force is the same as if we had a single cylinder at O, with a weight of reciprocating parts $= w$, and a crank set at an ange β ahead of the crank of the first cylinder.

Similarly the resultant couple has a moment the same as the moment about O, due to the inertia forces of a single cylinder with reciprocating weight $= w$ set at a distance l from O, and with a crank γ degrees in advance of that of the first cylinder.

It is interesting and instructive to trace the strains set up in the engine framing and upon the engine bearers by these inertia forces, but this is somewhat apart from the subject in hand. It

may be remarked, however, that the engine framing should be so formed, braced and tied as to act as a rigid truss in transmitting the inertia stresses to the structure of the ship. Otherwise local strains are liable to come upon the engine bearers which are extremely injurious.

TABLE I.

Name.	*Charleston.*	*Yorktown.*	*Vesuvius.*	*Cushing.*
No. cylinders of each engine	2	3	4	5
w_1 in pounds	6,800	1,800	860	210
w_2 "	10,000	2,200	1,010	210
w_3 "	...	3,000	1,080	210
w_4 "	...	...	1,080	210
w_5 "	..	...	...	210
l_1 in feet	0	0	0	0
l_2 "	6.75	3 6	3 5	4.
l_3 "	...	9.25	7.5	8.25
l_4 "	...	...	11.0	12.75
l_5 "	...	...	...	17.25
a_1	0	0	0	0
a_2	—90°	—120°	180°	144°
a_3	...	120°	—90°	—72°
a_4	...	...	90°	72°
a_5	...	...	...	—144°
$w_1\ l_1$	0	0	0	0
$w_2\ l_2$	67,500	7,920	3,535	840
$w_3\ l_3$	...	27,750	8,100	1,733
$w_4\ l_4$	...	...	11,880	2,678
$w_5\ l_5$	...	...	...	3,623
C = crank arm in feet	1.5	1.25	0.8333	0.625
r = revolutions per min.	115	160	280	370
.000341 $c\ r^2$	6.765	10.912	22.278	29.177
w (obtained from diagram)	12,100	1,060	148	0
l (obtained from diagram)	5.58	23.35	35.27	
$w\ l$	67,500	24,750	5,220	2,370

MEASURES OF AMELIORATION.

At one time it was a common practice to balance more or less completely the weight of the reciprocating parts of an engine by a weight fixed on the crank shaft opposite the crank.

It is easy to fix a weight thus so as to neutralize the inertia forces of the reciprocating parts, but new inertia forces equal to those destroyed will be introduced in a plane at right angles to the axes of the cylinders. It is impossible to balance a reciprocating weight by a rotating weight.

It will be remembered, however, that one-half the weight of each connecting rod was supposed to rotate uniformly. In addition we have for each cylinder the crank pin and arms rotating uniformly.

There are forces set up by the rotation of these weights whose amount can be easily determined by the preceding methods, but which are in any case small in comparison with the reciprocating forces. They can be practically neutralized by balance weights, and it appears desirable in the case of high-speed engines to neutralize them in this manner. The balance weight should never be greater than sufficient to balance the uniformly rotating weight.

While balance weights are nearly always fitted now for quick-running engines of any size, there are usually practical reasons for keeping them small, so that they are seldom as large as desirable. As has been stated, however, the forces due to uniformly rotating weights are comparatively small and negligible.

When we come to the forces due to the reciprocating parts we see that there are three sets of quantities more or less susceptible of change. These are:

1. The reciprocating weights themselves.
2. The distances between cylinder axes.
3. The crank angles.

The reciprocating weights for quick-running engines are usually already as light as practicable, so that the only change that can be made in them is to add to them. This is seldom desirable; for every increase of weight, while it may lessen the strain upon the engine bearers, must increase the local strains upon the engine framing proper.

The spacing of the cylinders is usually as close as practicable, so that the only change will be in the direction of an in-

crease. This has the disadvantage of making a longer engine, and at the same time is of but little benefit. The force polygon will undergo no change and, as a rule, it would require an impracticable change in the cylinder spacing to bring the last point (M_n) of the moment polygon much closer to the origin. With the crank angles now commonly used any increase in the cylinder spacing might be detrimental, as appears clearly from the diagrams for the *Yorktown* and *Vesuvius*.

We come now to the question of crank angles, and are met at once by the fact that the crank angles should be such as to make the turning moment on the shaft as uniform as possible. As a rule such angles are not those which give the minimum tendency to vibration. Still, in most cases, a slight change could be made for the better.

On the *Yorktown*, for instance, by setting the intermediate crank at an angle of $-87°$ with the high pressure instead of $-120°$, and the low pressure at $+133°$ instead of $+120°$, the force polygon $OP_1P_2P_3$ becomes a closed triangle and wl is reduced from 24,750 to about 22,000.

The turning moment on the shaft is, however, somewhat more irregular, so the net gain may be said to be very small.

It may also be slightly advantageous in certain cases to change the order of the cylinders, for instance in a triple-expansion three-cylinder engine to put the L.P. cylinder in the center. There are usually, however, good reasons for adopting a certain order of cylinders which outweigh any slight advantages as regards vibration of a different arrangement.

The true method which should be pursued to lessen the tendency of engines to cause vibration is to increase the number of cylinders. The facts as regards the *Yorktown* and *Vesuvius*, as shown in Table 1, point most clearly to this conclusion. The *Vesuvius* developed slightly more power than the *Yorktown*; her maximum reciprocating force was about 3,300 pounds and moment about the H.P. crank about 116,300 pound-feet. For the *Yorktown* the same quantities were 11,600 pounds and 270,000 pound-feet. The maximum reciprocating force of the *Vesuvius* was but little more than one-fourth, and the maximum

moment about two-fifths that of the *Yorktown*. A rough comparison of the actual engines of the *Vesuvius* with imaginary three-cylinder engines shows results still more unfavorable to three cylinders.

It may be safely concluded that for a small, high-powered, lightly-built ship an engine with at least four cylinders should always be adopted if vibration is to be avoided.

I shall now describe a method by which the tendency to vibration of a four-cylinder engine may be practically annulled, by the introduction of a suitable reciprocating weight.

It will be evident from what has gone before that in general, by suitably arranging the crank angles, the moment polygon of a four-cylinder engine at any point may be closed, *i. e.*, the moment reduced to zero. This means that the resultant action of the engine reduces to a reciprocating force through the point, and hence may be balanced by a reciprocating weight.

In general the moment polygon of a four-cylinder engine is a four-sided figure, except when the origin is in the axis of one of the cylinders, in which case one side of the polygon vanishes and it becomes a triangle.

So for any point away from a cylinder axis there are an indefinite number of crank-angle arrangements for which the moment polygon is closed. There can then be imposed a further condition. I shall select as this condition that the H.P. and L.P. cranks shall be at right angles, as tending, other things being equal, to produce a fairly uniform turning moment.

Taking now the case of the *Vesuvius* and determining the resultant force polygon when the moment polygon is closed for points forward of the engine, aft of the engine, and midway between each pair of cylinders, it is found that we cannot always close the moment polygon. The problem reduces itself to drawing a triangle with three sides known; but these sides are not always possible. The most suitable location for the reciprocating weight appears to be between the high and intermediate pressure cylinders. Increasing the distance between the H.P. and the I.P. cylinders to allow for the weight, the following appears to be about the best solution of the problem:

$l_1 = -3'\ 1\frac{1}{2}''$, $l_2 = 2'\ 8''$, $l_3 = 6'\ 8''$, $l_4 = 10'\ 2''$. Reciprocating weight = 1,325 pounds set at 230 degrees ahead of the H.P. crank.

The diagram, Fig. 6", shows the state of the case, and also that the L.P. cranks are opposite each other and perpendicular to the bisector of the angle between the H.P. and I.P. cranks. There are many cases where the introduction of a simple reciprocating weight as outlined above would be worth while; but at the same time it would be better still if this weight could be utilized. It seems to me perfectly practicable to place the air pump here, driving it by a crank and making its reciprocating parts sufficiently heavy.

Of course an air pump in this position would be a true air pump, *i. e.*, could pump only air and vapor from the condenser (which it would always be above); but there appear to be no serious difficulties in the way of fitting a reservoir beneath the condenser and a water pump which shall deliver water from this reservoir to the feed tanks as now done by the air pump.

I am told that such an arrangement was considered by the Bureau of Steam Engineering some years ago, but am not aware that it has ever been tried in practice.

I have discussed at some length the problem of the four-cylinder engine, for with triple expansion it is already desirable in many cases to divide the L.P. cylinder into two on the small high-powered ships, where vibration needs to be most considered. This is on account of the difficulty of getting the L.P. cylinder with necessary valves, &c., into the small space available on such ships. From the point of view of vibration the advantages of the four-cylinder arrangement, with properly spaced cranks, are incontestible.

With quadruple expansion it appears that for small high-powered ships we shall be driven to five cylinders, as on the *Cushing*. The methods previously described will enable us in such cases to annul vibration so far as it is due to the engines proper.

Take the case of the *Cushing* herself.

Fig. 6 shows two closed force and moment polygons (the moments are about the center crank).

Fig. 7 shows alternative arrangements of cranks corresponding to the closed polygons. It is evident that either arrangement is as good as the actual one as far as turning moment goes, and it is also evident that the tendency to vibration of the *Cushing*, so far as due to the engines proper, would be decidedly lessened if the same angles of cranks were retained but their order altered to correspond to either alternative of Fig. 7. By this arrangement the couple tending to produce vibrations would be reduced to about one-sixth of its present value.

It will probably have been observed that so far I have taken no account of valves and their gear in dealing with the reciprocating weights. This was purposely done, because I desired to avoid obscuring cardinal principles by a complication of details.

The position, weight, stroke and angular advance of each valve is commonly fixed for each cylinder. While it would be possible to draw the force and moment polygons with sides corresponding to each valve as well as cylinder, it is preferable to use resultant sides, so to speak, corresponding to each cylinder.

Thus suppose the third cylinder has two valves with an angular advance $= a$. If $w'_3 w_3''$ are the weights of the valves, $l' l''$ the distances of their centers from the origin, and c'_3 their stroke, we shall evidently have, instead of $w_3 c \cos a_3$, $w_3 c \cos a_3 + w'_3 c'_3 \cos (a_3 + a_3) + w''_3 c'_3 \cos (a_3 + a_3)$. Then (see Fig. 8) draw $P_2 P'_3 = w_3 c_3$, $P'_3 p' = w'_3 c'_3$ and inclined at an angle a to $P_2 P'_3$, and $p' P_3 = w''_3 c'_3$ and inclined at an angle a to $P_2 P'_3$. Then $P_2 P_3$ is the new side of the force polygon, and after using it, as previously described, the crank angles are readily obtained by reversing the process just described. It will be observed that it is necessary in this method to introduce the stroke of the parts as well as their weights. This may be avoided by multiplying the weight of each valve by the ratio between its stroke and the stroke of the engines. The method to be used in determining the resultant side of the moment polygon to be used for each cylinder is obvious from the preceding.

As a rule, by taking account of the valves, there is not much

change made in the crank angles necessary to render both the force and moment polygons closed.

In concluding what I have to say about forces developed in the engine room, I wish to point out clearly that the preceding results were obtained by neglecting connecting rod obliquity, slackness of brasses and fastenings, and flexibility of engine framing. The results obtained are then only approximations; but there can be little doubt about their being very close approximations. I would here say that from the point of view of vibration it is scarcely possible to make the engine framing too rigid.

I come now to the second branch of my subject, namely, a discussion of the tendency to vibration due to forces called into play by the action of the propeller.

When a symmetrical propeller is advancing steadily, parallel to its axis, with a constant turning moment, through water which, when undisturbed by the propeller, has no motion unless it be a uniform one parallel to the axis of the propeller, the resultant of the reactions upon the propeller must be forces constant in direction and magnitude, or else constant turning couples.

Hence there will be no tendency to vibration due to the action of such a propeller, and it will be necessary to examine wherein the conditions of working of actual propellers differ from the imaginary conditions above:

1. Lack of symmetry.

The two, three and four-bladed propellers used in practice are always intended to be symmetrical, their departure from symmetry being due to accidental causes and faulty workmanship. In examples of the best practice the departure from symmetry is small.

2. Non-uniformity of turning moment.

This always exists in practice.

3. Axis not parallel to direction of advance.

This is nearly always the case as regards vertical direction, since the shaft is parallel to the water line for one trim only. Lateral deviations of the shaft from a fore-and-aft direction

occur only in twin-screw ships. In addition, the axis of the propeller may not coincide exactly with the axis of the shaft.

4. Non-uniformity and lack of parallelism of the race.

These conditions are always present. The following principal causes for their existence may be stated:

(a) The water closing in around the stern has a tendency to follow the lines of the ship and hence make an angle with the direction of advance.

(b) Owing to stream-line action toward the stern, the water close to the side tends to flow aft more slowly than the water which is more remote.

(c) Owing to the "frictional wake," or following current, caused by the friction of the skin of the ship, the uniformity of the race is further disturbed. The frictional wake is strongest close to the side and near the surface, and diminishes as we go out and down.

(d) The formation of surface waves by the ship further disturbs the race. The disturbance is greatest at and near the surface, and varies in direction as the propeller falls beneath a crest or a hollow.

I shall first investigate the case of a propeller advancing in undisturbed water but with its axis inclined at an angle to the direction of advance. The method adopted by Froude in determining thrust &c. of a propeller is most suitable for my purposes. Referring to Fig. 9, let LL denote a small element of a propeller blade at distance r from the axis AA.

Then if MN = pitch of the element, and $OM = 2\pi r$, the figure may also be taken as a diagram of velocities. MN being speed of advance with no slip, MS the actual speed of advance, $SON = \varphi$, (called the slip angle) and OS the actual velocity of the element in direction and magnitude. Now for a small plane of area dA, Froude determined that the thrust in the direction of the axis, denoted by dT, is equal to:—A constant $\times\ \varphi\ \times\ dA\ \times$ a quantity depending on the velocity and radius of the element.

Now the angle φ varies from point to point, and there is some inaccuracy in applying this formula for a small isolated plane to a small element of a continuous blade. The formula is, how-

ever, sufficiently accurate for our purpose. If we denote by dS the tangential forces acting on an element, we have:

$$dT = k_1 \varphi \, dA.$$
$$dS = k_2 \varphi \, dA.$$

k_1 and k_2 being constants.

Suppose now (Fig. 10) we incline the axis of the propeller through a small angle δ in a horizontal plane. Then if a denote the angular distance of the radius of the element from the vertical, the difference between the inclinations of the undisturbed and the disturbed elements LL $L'L'$ to the direction of the advance (*i. e.*, that of the undisturbed axis) will be, to the first order of small quantities, $\delta \cos a$.

The expressions for thrust &c., are changed to—

$$dT = k_1 (\varphi - \delta \cos a) \, dA.$$
$$dS = k_2 (\varphi - \delta \cos a) \, dA.$$

Suppose now under these conditions we take the sum of all the elementary forces for the whole of one blade. We shall have a single resultant force whose fore-and-aft component I shall denote by T, and tangential component by S, while the distance of its point of application from the axis I shall denote by r_0. Then the turning moment $= Sr_0$. S and T are measured in pounds, r_0 in feet.

Now, further, let φ_0 denote the average value of φ along the blade. Table II gives values of φ_0 for various pitch ratios and real slips, and a standard diameter of hub $= \frac{1}{4}$ the diameter of the propeller. This is a sufficiently close approximation to good man-of-war practice.

TABLE II.—VALUES OF φ_0.

Slip per cent.	5	10	15	20	25	30	35	40
Pitch ratio.								
1.0	1°09′	2°22′	3°35′	4°52′	6°11′	7°32′	8°53′	10°22′
1.2	1°15′	2°35′	3°54′	5°17′	6°43′	8°11′	9°43′	11°16′
1.4	1°19′	2°43′	4°06′	5°35′	7°07′	8°42′	10°23′	12°07′
1.6	1°22′	2°48′	4°15′	5°47′	7°24′	9°03′	10°49′	12°40′
1.8	1°23′	2°50′	4°19′	5°53′	7°34′	9°18′	11°08′	13°03′
2.0	1°23′	2°50′	4°21′	5°57′	7°39′	9°25′	11°19′	13°13′

Then for a given propeller under given conditions we may write:

$$T = K_1 (\varphi_0 - \delta \cos a),$$
$$S = K_2 (\varphi_0 - \delta \cos a).$$

K_1 and K_2 are constants depending upon the conditions, and will be discussed later.

Now let H be the horizontal component of S, being that component which would in an actual case act parallel to the water and athwartships.

Also let V denote the vertical component of S. Then evidently:

$$H = S \cos a,$$
$$V = S \sin a.$$

It is necessary here to be careful of our signs. In a twin-screw ship with shaft turning outward, and a measured from the highest position of a blade, H, when positive, will denote a force acting inboard, and V, when positive, a force acting upward.

Also a positive value of δ will denote an inclination outward of the shaft—looking at it from behind.

Then for a single blade:

$$T = K_1 (\varphi_0 - \delta \cos a),$$
$$S = K_2 (\varphi_0 - \delta \cos a),$$
$$H = S \cos a = K_2 (\varphi_0 \cos a - \delta \cos^2 a),$$
$$V = S \sin a = K_2 (\varphi_0 \sin a - \delta \sin a \cos a).$$

The propellers of practice, however, have two, three or four blades. Suppose then we have a propeller of n blades, set at angles $a_1, a_2 \ldots a_n$, with a fixed radius whose angular distance from the vertical is denoted by a. Our expressions become:

$$T = K_1 [n \varphi_0 - \delta \{\cos (a + a_1) + \cos (a + a_2) + \ldots + \cos (a + a_n)\}],$$
$$S = K_2 [n \varphi_0 - \delta \{\cos (a + a_1) + \cos (a + a_2) + \ldots + \cos (a + a_n)\}],$$
$$H = K_2 [\varphi_0 \{\cos (a + a_1) + \cos (a + a_2) + \ldots + \cos (a + a_n)\} - \delta \{\cos^2 (a + a_1) + \cos^2 (a + a_2) + \ldots + \cos^2 (a + a_n)\}],$$

$$V = K_2 [\varphi_0 \{\sin(a + a_1) + \ldots + \sin(a + a_n)\} - \delta \{\sin(a + a_1)\cos(a + a_1) + \sin(a + a_2)\cos(a + a_2) + \ldots + \sin(a + a_n)\cos(a + a_n)\}].$$

These expressions are thoroughly general. In applying to actual cases it is desirable to have a correspond with the mean radius of one blade. Then $a_1 = 0$ and $a_2\ a_3 \ldots$ are the angular distances of the second, third, &c., blades from the first.

Then for two blades:

$$a_1 = 0 \qquad a_2 = 180$$
$$T = 2 K_1 \varphi_0,$$
$$S = 2 K_2 \varphi_0,$$
$$H = -2 K_2 \delta \cos^2 a,$$
$$S = -K_2 \delta \sin^2 a.$$

For three blades:

$$a_1 = O \qquad a_2 = 120 \qquad a_3 = -120$$
$$T = 3 K_1 \varphi_0,$$
$$S = 3 K_2 \varphi_0,$$
$$H = -3/2\ K_2 \delta,$$
$$V = 0.$$

For four blades:

$$a_1 = 0 \qquad a_2 = 90 \qquad a_3 = 180 \qquad a_4 = -90$$
$$T = 4 K_1 \varphi_0,$$
$$S = 4 K_2 \varphi_0,$$
$$H = -2 K_2 \delta,$$
$$V = 0.$$

These results show that a slight inclination of the axis of a propeller produces no change in the thrust or turning moment. It should be observed that such an inclination varies the slip of a blade from point to point and so produces a result analogous to a propeller working in a current where the velocity varies from point to point, as in the frictional wake of a vessel.

Now, Froude's experiments have shown that in such a case the thrust and turning moment of a propeller are practically unchanged, provided the mean slip and speed of advance are unchanged. So the conclusions deduced from these formulæ,

as regards constancy of thrust and turning moment, may be said to be borne out by experiment.

Now as to H and V.

It appears that for a two-bladed propeller, by turning the shaft slightly out as we go forward, we introduce a horizontal force which always acts out, and varies from zero when the blades are horizontal to $2\ K\ \delta$ when the blades are vertical.

There is also introduced a reciprocating vertical force which is at a maximum when the blades are at 45° and is zero when they are either vertical or horizontal.

For three or four-bladed propellers, by turning the shaft out, we introduce a constant horizontal outward force equal in amount to one-half the constant turning force multiplied by the ratio between the deflection δ of the shaft and the mean slip angle.

These conclusions appear novel, but are, I believe, borne out by experience. It is a well recognized fact that two-bladed propellers, other things being equal, cause greater vibration than three or four-bladed, and three or four-bladed propellers are usually considered equally good as regards vibration.

As regards the direction of the outward force it has been pointed out by Thorneycroft and Barnaby that the action of the screw tends to steady a ship on her course. The expressions just deduced show that if the shaft, and head of the ship, are deflected to starboard there is developed a horizontal force tending to pull the propeller, and stern of the ship, to starboard, and hence to steady the ship.

It is now advisable to form some idea of the amount of the forces I have been discussing. Let the horse-power delivered to the propeller be taken as a certain fraction, say q, of the indicated horse-power I. Then $q\ I$ is the horse-power delivered to the propeller. Let r = the revolutions, p = pitch of propeller, and d = diameter of the propeller.

It will be remembered that r_0 was taken as the radial distance of the point of application of the resultant transverse force on a blade.

Then the work done in a single revolution $= 2 \pi r_0 S$ foot-pounds, and in one minute $= 2 \pi r_0 S \times r$ foot-pounds. But this must be the work delivered to the propeller in foot-pounds $= q I \times 33000$.

Then $33000 q I = 2 \pi r_0 S \times r$.

Let n = number of blades.

Then $S = n K_2 \varphi_0$.

Whence:

$$33000 q I = 2 \pi r_0 r \times n K_2 \varphi_0.$$

Now r_0 is somewhat uncertain, depending on shape of blades &c., but may be said to be always between $\frac{2}{3}$ and $\frac{3}{4}$ of the radius of the propeller. Then $2 r_0 = .7 d$ will be a fair approximation.

Then:

$$K_2 \varphi_0 = \frac{33000 q I}{.7 \pi d r n}.$$

Now q will vary from .8 to .9, and may be reasonably taken as .85. Then reducing,

$$K_2 \varphi_0 = \frac{12750 I}{n d r}.$$

Now an average good propeller efficiency is 66 per cent. If s = the slip per cent., $T \times p \times (1 - s) \times r$ = useful work per minute $= \frac{2}{3}$ (work delivered to the propeller) $= \frac{2}{3} \times 33000 \times q I$.

Whence:

$$n K_1 \varphi_0 p (1 - s) r = 22000 q I.$$

or,

$$K_1 \varphi_0 (1 - s) = \frac{18700 I}{n p r}.$$

It appears then that a knowledge of the actual slip in any case is essential to a knowledge of K_1 and K_2.

The actual slip in any case is a somewhat difficult thing to obtain, since the exact speed of the frictional wake is not known. For our purposes, however, we may take the average speed of wake as 10 per cent. of the speed of the ship. Then Table III, below, shows the necessary particulars and values of K_1 and K_2 for the *Yorktown* and *Vesuvius*.

TABLE III.

Name.	*Vesuvius.*	*Yorktown.*
I. H. P. (one engine)	2,150	1,700
Speed knots	21.6	16.65
Diameter of propeller = d	7.75	10.5
Pitch of propeller = p	9.375	12.5
Number of blades	3	3
Revolutions	279.0	160.8
Approximate true slip, per cent.	24.7	24.5
Value of pitch ratio	1.21	1.19
Value of φ_0 (Table II)	6.75	6.70
Value of K_1	1,008	1,056
Value of K_2	626	639

It is evident from the preceding that a deviation of the shaft axis will produce a strain upon the bearing which carries the propeller.

Now a motion of the water such as to make the slip at a point of a blade vary during a revolution amounts to a virtual deviation.

The closing in of the water around the stern of a ship affects the slip at various points of the revolution; but its effect would seem to be comparatively small in most cases. In my opinion the principal cause of disturbance is the fact that the velocity of the water of the race is by no means uniform, owing to the fact that the propeller works in the frictional wake.

In a single-screw ship, for instance, the slip must be greater than the average when a blade is about its highest position, and less than the average when the blade is toward the bottom. For a right-handed screw this amounts to a virtual deflection of the shaft to port, and for a left-handed screw to starboard. This fact is well known, and several people have suggested the use of feathering blades to equalize the slips at various points of the revolution, and Mr. Thorneycroft has stated that he had thought of inclining the shaft in a horizontal line.

The amount of virtual deflection, and hence of sideward thrust, due to non-uniformity of race can be determined only in the light of fuller knowledge of the frictional wake and stream

line action at the stern. Many experiments on the subject have been made by Froude, but little has been published.

We do know, however, that under ordinary working conditions of a twin-screw propeller, the slip is greater inboard and above the hub than outboard and beneath the hub.

For a propeller turning out this amounts to a virtual deflection down and in, starting from the propeller, and should be met by deflecting the shaft up and out.

Suppose, for instance, that a propeller turning out with pitch ratio 1.4 has an average slip of 20 per cent., a slip of 25 per cent. for the extreme inboard and upward positions and of 15 per cent. for the extreme outboard and downward positions. From Table II it is seen that a slip of 15 per cent. corresponds to $\varphi_0 = 4° 06'$; 20 per cent. to $5° 35'$, and 25 per cent. to $7° 07'$. Evidently then, if the shaft were inclined $1° 30'$ up and the same amount out, there would be much greater uniformity of action.

As regards vibration of the ship, it appears from the preceding that a moderate amount of deflection—real or virtual—is not a very serious matter.

It is, however, decidedly objectionable as increasing the strain on hangers &c., and on the propeller blades and hub, owing to the violent fluctuations of stress. There must also be some loss of efficiency.

In this connection it may be remarked that there are sometimes reasons for turning a propeller shaft in a certain way, out or in, up or down. On a twin-screw ship a shaft which leads out and up from the propeller should revolve outboard; one which leads in and down should revolve inboard if the maximum uniformity of action is desired.

With horizontal engines it is objectionable to turn inboard, because then the pressure on the crosshead guides is upward.

With vertical engines, however, it is frequently immaterial in which direction the crosshead pressure acts—whether outboard or inboard—and the direction of rotation can be chosen which best suits the other conditions.

The bearing of the above upon the question of vibrations is this: If a shaft with an actual deviation in a certain direction is

turned in such a manner as to give it a virtual deviation in the same direction, it is practically certain that the methods previously used, which are suited only for small angles of deviation, would not apply exactly for the large deviations thus obtained. The horizontal force would not be constant, nor the vertical force always zero. Thus when small boats are turning rapidly their propellers are working under conditions of large virtual deviation, and there is nearly always an increase of vibration.

There remains one matter to be considered as regards tendency of propellers to cause vibration, namely the inequality of the turning moment. This is not so great at the propellers as at the engine, owing to the fly-wheel effects of the moving parts and the elasticity of the shafting. It appears probable, however, that it takes effect almost entirely in changing φ_0 and produces little change in the constants K_1, K_2. The principal result would then be a variation of the thrust. While undesirable, it does not appear probable that this is a very serious cause of vibrations.

Summing up, I think it may be reasonably concluded that in ships with three or four-bladed propellers the direct forces exerted by the propeller and tending to cause vibrations are small. Being at the extreme end of the ship, however, they have a very powerful leverage and may produce results much greater than would be expected. It appears then very desirable so to arrange the axis of the shaft and direction of revolution as to bring the propeller as nearly as possible to the condition of working in a uniform race.

Table II affords a means of approximating to this result. In using it, it may be assumed that the wake has an average speed of 10 per cent. of that of the ship, the maximum being 15 per cent. and the minimum 5 per cent. These seem reasonable allowances, but further experiment upon the subject is exceedingly desirable.

It should be pointed out that the propeller may have a very serious indirect influence upon vibrations. It is well known that certain speeds of revolution of the engines are liable to cause excessive vibration. This is because the period of revolu-

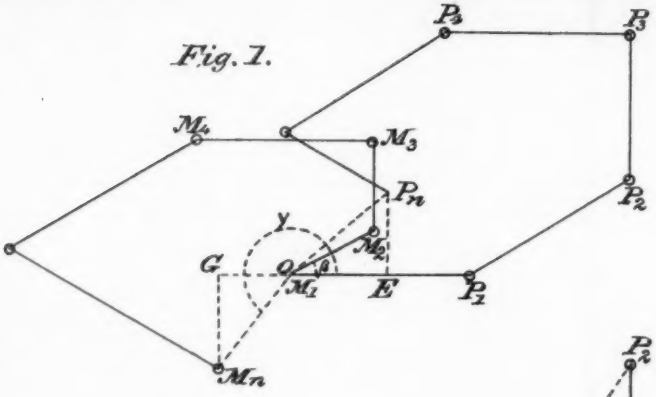


Fig. 1.

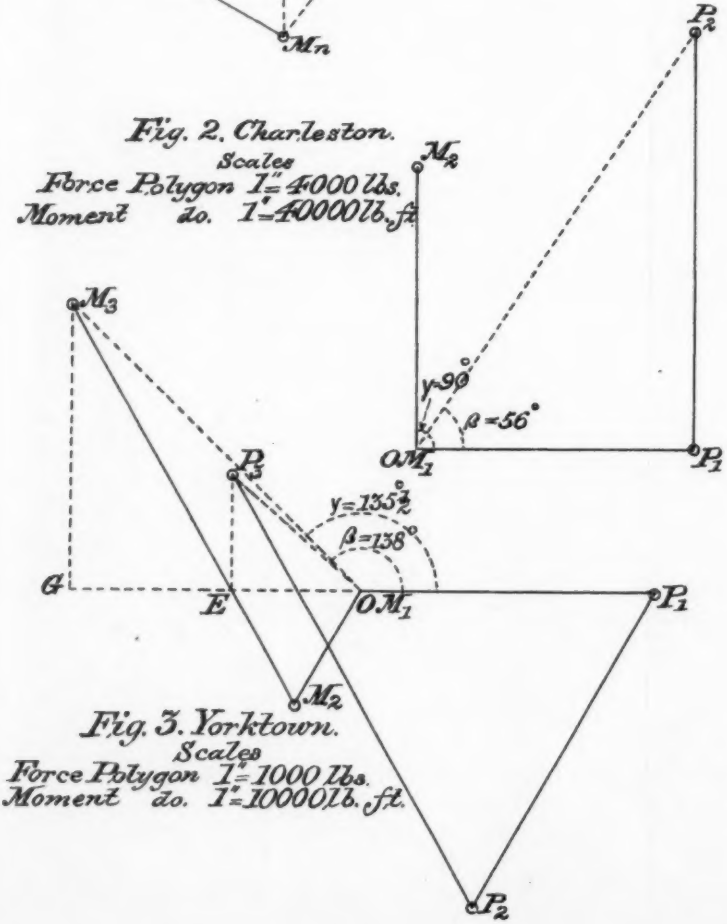
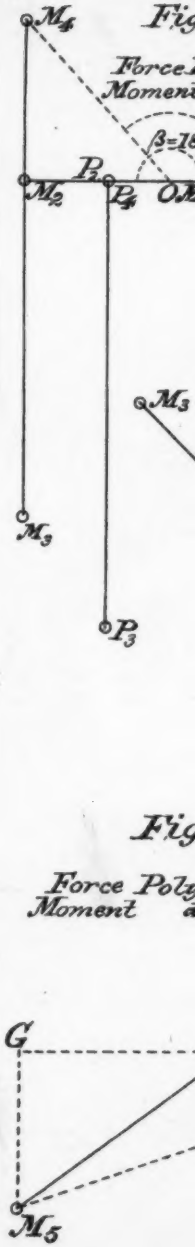


Fig. 2. Charleston.
Scales
Force Polygon 1"=4000 lbs.
Moment do. 1"=400000 lb. ft.

Fig. 3. Yorktown.
Scales
Force Polygon 1"=1000 lbs.
Moment do. 1"=100000 lb. ft.



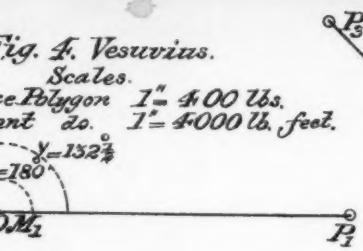


Fig. 4. Vesuvius.
Scales.
Polygon 1" = 400 lbs.
do. 1" = 4000 lb. feet.
y=132½°
=180°
M1
P1
P3
P2
P4

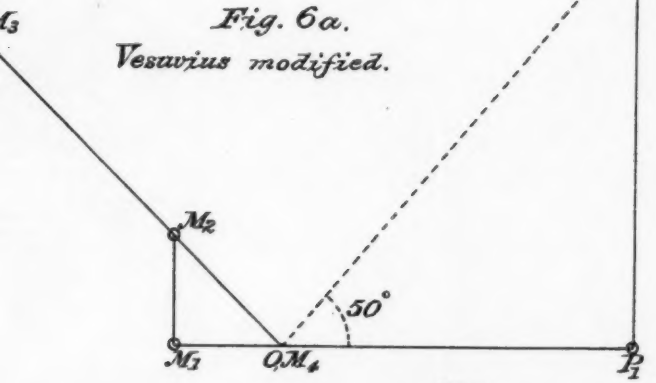


Fig. 6a.
Vesuvius modified.
M3
M2
50°
M1
O,M4
P1

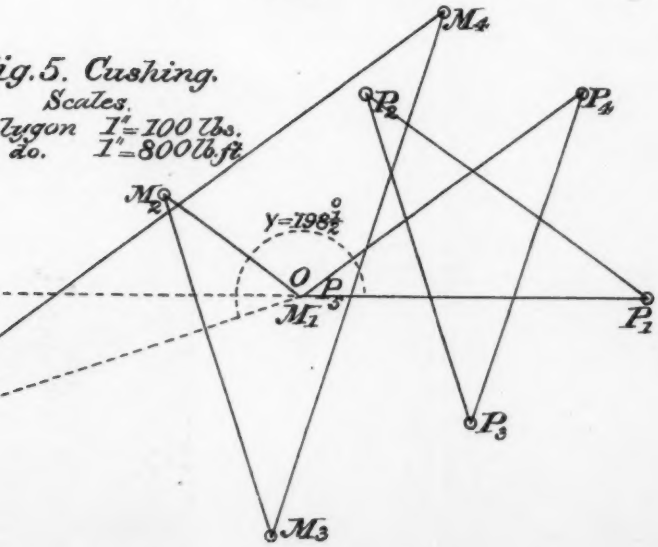


Fig. 5. Cushing.
Scales.
Polygon 1" = 100 lbs.
do. 1" = 800 lb. ft.
M4
P2
P4
M2
y=198½°
O
P3
M1
P1
P3
M3

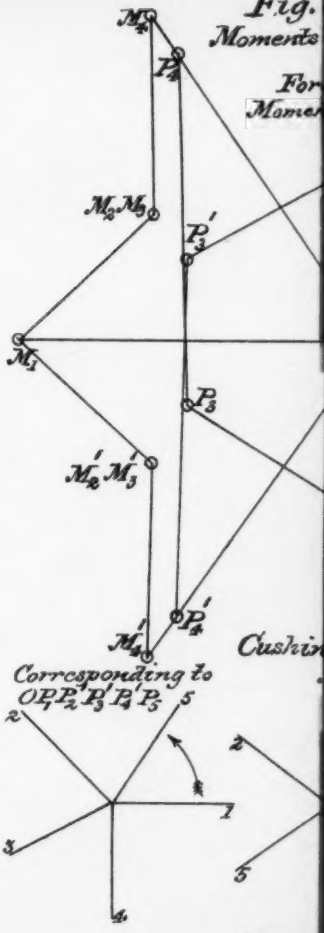


Fig.
Moments
For
Momen
M4
P4
M2,M3
P3'
M1
P3
M2' M3'
P4'
M4'
Cushin
Corresponding to
OP1P2P3P4P5
5
2
1
3
4
2
5

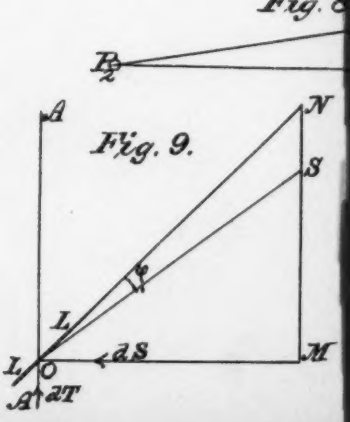


Fig. 8
P2
A
N
Fig. 9.
S
L
dS
M
L
O
A
dT

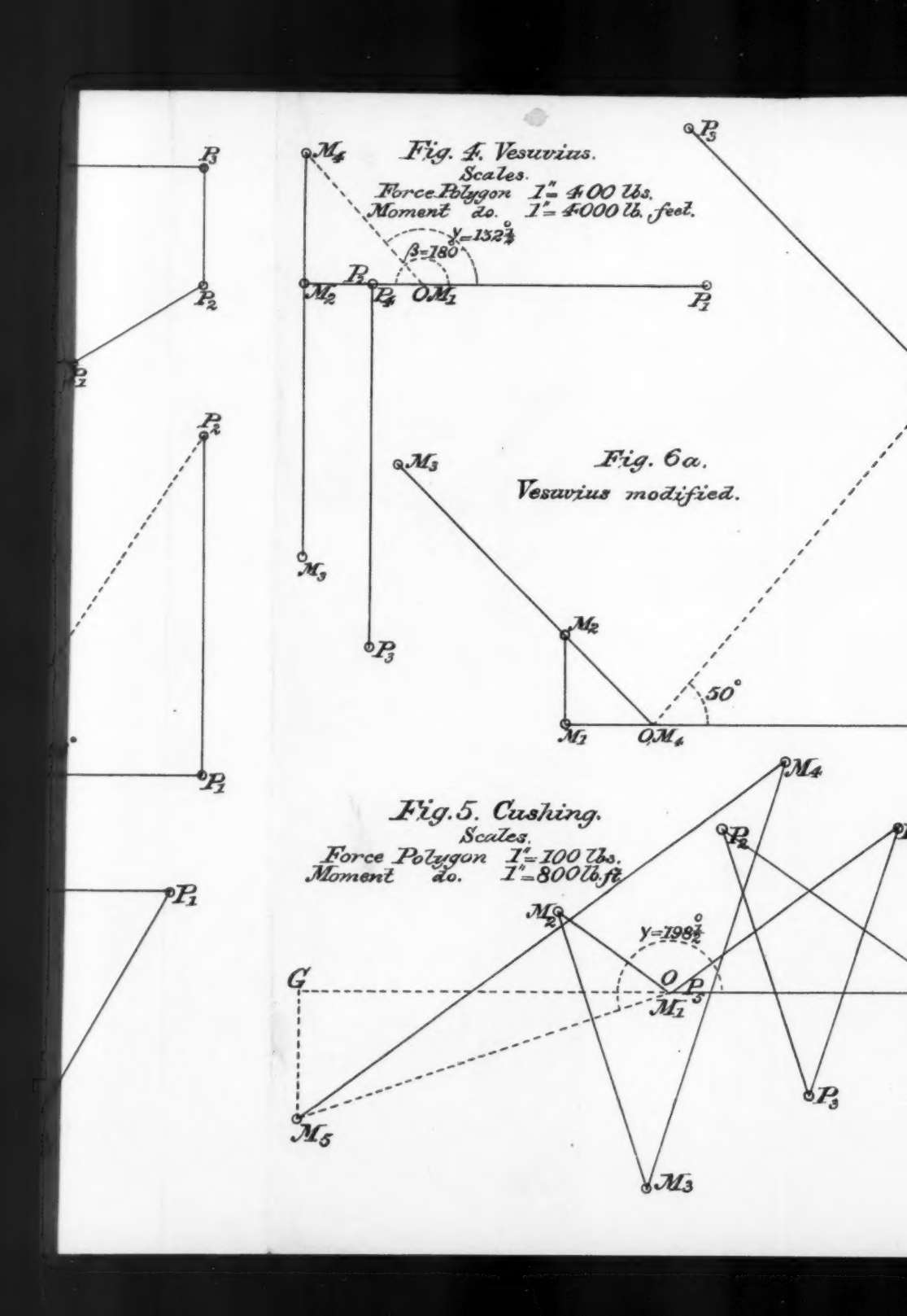


Fig. 4. Vesuvius.
Scales.
Force Polygon 1" = 400 lbs.
Moment do. 1" = 4000 lb. feet.
γ=132½°
β=180°
M4
P3
P2
M2
P4
O
M1
P1
Fig. 6a.
Vesuvius modified.
M3
M2
50°
M1
O
M4
Fig. 5. Cushing.
Scales.
Force Polygon 1" = 100 lbs.
Moment do. 1" = 800 lb ft
M4
P2
M2
γ=198½°
G
O
P3
M1
P3
M5
M3

Fig. 6. Cushing modified.
Moments about axis of central cylinder.
Scales
Force Polygons 1" = 100 lbs.
Moment do. 1" = 200 lb. ft.

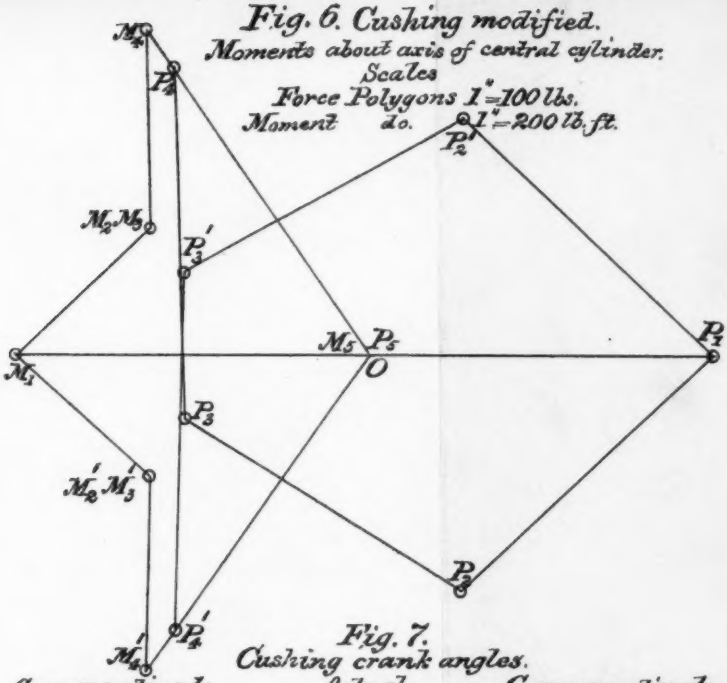


Fig. 7.
Cushing crank angles.

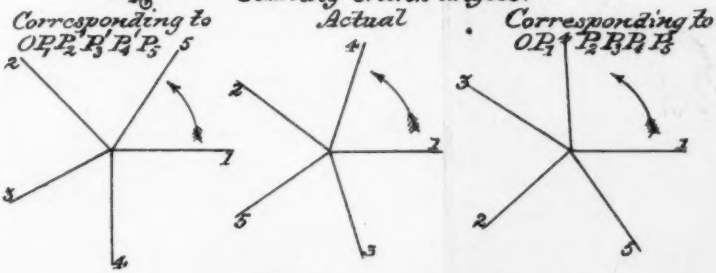


Fig. 8.

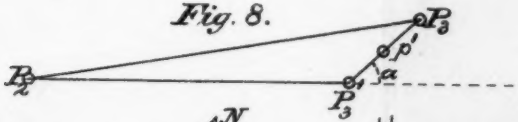


Fig. 9.

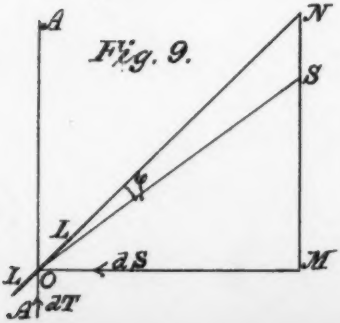
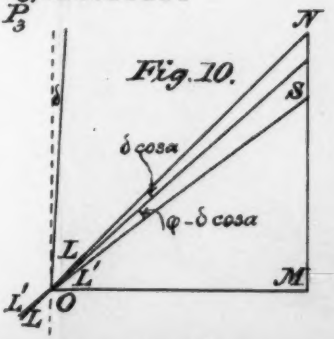


Fig. 10.



tion of the engines synchronizes with the period of vibration of the ship—the engines and ship may be said to be "in step."

By changing the propeller we can, in many cases, change the revolutions of the engines and so cause them to "break step," so to speak, with the ship. This has been frequently done, and in some cases a great many different propellers have been tried.

Nearly all cases of excessive vibration of properly designed ships are due to synchronism, and it would evidently be very desirable to be able to determine beforehand the period of vibration of a given ship. Any method for such a purpose must necessarily be founded upon numerous experiments with various types of ships. So far as I am aware experiments upon this subject have been confined to foreign nations, and very little data of value ever been published.

IV.

AN INVESTIGATION OF ASSISTANT ENGINEER W. H. ALLDERDICE'S "NOTES ON ANALYSIS OF ENGINE TRIALS."

By Chief Engineer Isherwood, U. S. Navy.

On page 510 of the last issue (November, 1890) of The Journal of the American Society of Naval Engineers, Assistant Engineer Allderdice, U. S. Navy, objects to the methods I have adopted for computing the liquefaction of steam when working expansively; for determining the total horses-power developed by steam in the cylinder; for the distribution of that total horses-power; and for calculating the thrust of propelling screws. As he has given some reasons for his objections, and as the subjects, from an engineering point of view, are interesting and important, I propose in this paper to consider them and to show that the methods in question are correct, that the rea-

sons assigned against them are baseless, and that the methods recommended in place of them are erroneous.

First, with regard to the method of computing the liquefaction of working steam during expansion. Mr. Allderdice begins with the statement that "the total condensation of steam in the cylinder of an engine using steam expansively may be considered as composed of two parts, the first and usually the greater part being that which is due to the cooling effect of the metallic walls of the cylinder; the second, that which is due to the loss of heat by the expanding steam while doing work, after communication with the source of heat has been shut off. In a non-conducting cylinder the second part only would exist."

The above citation is quite erroneous in limiting the liquefaction of saturated steam working expansively in a cylinder to the two causes stated. Several other causes combine to produce this liquefaction, and some of them are important factors in the present problem. Indeed that problem cannot be completely conceived without them, and yet they have been ignored not only by Mr. Allderdice, but by all others except myself who have attempted its solution.

Premising that the boiler supplies only dry saturated steam to the steam pipe at the boiler end, and that this steam loses no heat by radiation while in the pipe, the steam delivered by the pipe to the cylinder will nevertheless be *wet* steam, by which term is meant steam whose every separate molecule has undergone liquefaction in part. It means the molecular liquefaction caused by internal work, as distinguished from the mass liquefaction caused by external radiation, or by contact with external cold. Molecular liquefaction takes place on every molecule of the steam, while mass liquefaction takes place at only the boundaries of the steam mass. The latter produces water in sensible quantity on the boundary surfaces of the steam. The former produces water in quantities too minutely divided to be sensible, and these comminuted quantities are as intimately intermingled with the steam as are its own molecules, giving it the appearance of fog, cloud, mist or haze. Now, there can be no internal or molecular work done by saturated steam that is not accompanied

and evidenced by this molecular liquefaction, and internal or molecular work can be done by steam only when it is expanded, in which case molecular work is always done by it whether or not the steam during its expansion does external or mass work. The steam may expand without its communication with the boiler being cut off, as occurs in the steam pipe conveying steam from the boiler to the cylinder during the portion of the stroke of the piston preceding the closing of the cut-off valve, or it may expand after its communication with the boiler has been cut off, as occurs in the cylinder after the closing of the cut-off valve; but there must always be expansion of the steam—that is the only condition. Unexpanded steam is perfectly transparent and invisible; it cannot do internal work, and the heat for any external work that it may do is obtained directly from the furnace of the boiler.

The boiler furnishes steam of the boiler pressure to the boiler end of the steam pipe, but from that end to the cylinder, throughout the entire length of the pipe, the steam is delivered by its own expansion, its pressure at the cylinder end of the pipe being less than at the boiler end. This is the only way in which elastic fluids ever move. The steam is not pushed through the pipe like a solid block by the boiler pressure. It expands in the pipe, reacting against the boiler pressure and propelling itself forward by its elasticity in virtue of that reaction. The curve of lessening pressures in the pipe, from the boiler to the cylinder, will approximate the hyperbola. The longer the pipe, other things equal, the greater will be the difference of the pressure at its two ends; the greater the cross area of the pipe, other things equal, the less will be the difference of the pressure at its two ends; and the denser the boiler steam, other things equal, the greater will be the difference of the pressure at the two ends of the pipe. The curve of lessening pressures in the steam pipe is not sensibly affected within the limits of ordinary practice by the fact of that pipe being covered with non-heat-conducting substances, or being bare. Such covering decreases the surface condensation of the steam, thereby increasing the economy of the fuel, but does not influence the law of expansion of the steam, nor, excepting

to a trifling degree, the resulting pressures, the latter being always measured from the zero line.

Now the expansion of the steam in the pipe is accompanied by the performance of both mass work and molecular work, the result of which is a liquefaction of so much of that steam as may be required to liberate the heat transmuted into such work. The mass work is the transportation during a given time of the steam from the boiler to the cylinder, and is measured by the difference between the bulk of the steam at the boiler pressure and its bulk at the mean pressure in the cylinder previous to the closing of the cut-off valve, multiplied by that mean pressure. Expressing in cubic feet, the difference of the two bulks of the same weight of steam used during the given time, and expressing the above mean pressure in pounds per square foot, the product will be the work done in foot-pounds, dividing which by the Fahrenheit mechanical equivalent of heat there results the number of Fahrenheit units of heat transmuted into this mass work. But more heat than this has to be taken from the steam, namely, the heat transmuted into the internal or molecular work of the steam during its expansion in the pipe. In the case of an ideal gas there would be no such work, but a vapor is vastly removed from that condition, and cannot be expanded without the expenditure of internal work upon its molecules to overcome its inter-molecular forces. What is called the latent heat of steam is the quantity of heat required to maintain unit of weight in the condition of steam under the given pressure. This quantity, for equal weights of steam, decreases as the pressure increases, because the temperature increases with the pressure. Now the effect of expanding steam, in a working cylinder, is to decrease its pressure and, consequently, correspondingly increase its latent heat. The difference of the latent heats before and after expansion, expresses the quantity of heat that must be added to a given weight of steam to maintain it in the state of steam after expansion, and this quantity is obtained from the steam itself by a portion undergoing liquefaction in order to free the quantity of heat required to maintain the remainder in the form of steam. The fraction of the weight of steam so liquefied, will

be the quotient of the division of the difference between the two latent heats by the latent heat before expansion.

The expansion of steam, with or without doing external work, is accompanied by a fall of temperature. There is less sensible heat in the unit of weight of steam after expansion than before, and, assuming that throughout the expansion the steam remained saturated, having the temperature normal to the pressure, the sensible heat liberated by the expansion would be expressed per unit of weight, by the product of the specific heat multiplied by the difference of temperature, the latter being always measured from the absolute zero. As the apparent specific heat of saturated steam is at present unknown, this calculation cannot be made, nor would it be of any use in the present problem if it could, for its effect is included in the latent heats of the steam at the two pressures. The latent heat of steam is the *total* quantity of heat in it, this heat representing the *vis viva* of the steam molecules due to any cause whatever. To prove this fact, which seems to have been completely overlooked by previous investigators, there is only to be considered the manner in which the quantities called latent heat were obtained by Regnault. He drew from a boiler a pound weight of saturated steam of a given pressure, and he liquefied it under the same pressure by means of water of known temperature: now the number of pounds of this water multiplied by the difference between the units of heat in it before and after the liquefaction of the steam, is the latent heat of the pound of steam. As in the case of both pressures the temperature of the water of liquefaction is the temperature of the steam before liquefaction, the effect of the lowering of the temperature from that which is normal to the higher pressure to that which is normal to the lower pressure, is included in the latent heat of the latter.

From the foregoing will be perceived that, supposing the steam room of the boiler to contain perfectly dry saturated steam, the cylinder will always be supplied with *wet* steam—that is, with steam intermingled with finely comminuted water—each molecule of steam being associated with a molecule of water. There is no method by which dry steam can be supplied to the

cylinder except superheating the steam previous to its entrance there. Yet this condition of the steam appears to have been always ignored and a miracle supposed to exist, namely, that the transportation of the steam from the boiler to the cylinder was accomplished without the expenditure of work, and, consequently, without the expenditure of heat. In other words, that something was obtained for nothing; but nature does not indulge in any such generosity. She inflexibly exacts the full equivalent of whatever she bestows. She exchanges equivalents, but gives nothing.

There follows from the preceding that the steam *always* expands more or less in the cylinder from the commencement of the stroke of the piston to the point in that stroke at which the cut-off valve closes, as well as after that point; and as a result of this expansive working, which may be little or much according to the proportions of the mechanism and regimen of the steam, it will undergo whatever liquefaction may be needed to furnish the heat transmuted into the power, great or small, produced by this expansion; and such liquefaction should be added to the other liquefactions. These facts have never before been noticed, but they have only to be stated to command assent.

After the closing of the cut-off valve the steam in the cylinder expands uninterruptedly to the end of the stroke of the piston. There thus appears that the steam really expands from the boiler end of the steam pipe to the end of the stroke of the piston, only under different conditions before and after the closing of the valve. Before the closing the expanding steam remained in communication with the boiler steam; after the closing it had no communication with the latter. All previous investigators, however, are silent as regards these facts and their effects; yet the continuous expansion of the steam in the steam pipe can be easily shown experimentally by drilling holes at intervals along the top and inserting pressure gauges in them, when a regular fall of pressure will be found from the boiler to the cylinder, and sensibly the same whether the pipe be bare or whether it be covered with non-heat-conducting material. To the mass and molecular work done by the expansion of the steam in the steam

pipe, is owing the considerable loss of superheating experienced in the passage of superheated steam from the boiler to the cylinder. Under the conditions of ordinary practice from twenty to thirty degrees Fahrenheit of temperature are thus lost, which loss is the thermal equivalent of the mass and molecular work done by the expansion of the steam during its transit from the boiler to the cylinder. It is not the result of external radiation, for it takes place with the pipe so completely swaddled with non-heat-conducting material as to practically prevent radiation. It represents the heat transmuted into work.

From the preceding appears that a portion of the work done by the steam between the commencement of the stroke of the piston and the closing of the cut-off valve, has been done by steam expanding between its pressures at the points just named. Of course, there is a liquefaction of steam corresponding to the work of such expansion which should be added to the other liquefactions. In fact, the steam is used expansively always, from the commencement to the end of the stroke of the piston, only very much more expansively after the closing of the cut-off valve than before. Finally, when the end of the stroke of the piston is reached and the exhaust port opens, the steam is pushed out of the cylinder by its own expansion. It is not pushed out, strictly speaking, by the piston during the return stroke; the piston only compresses the back pressure sufficiently above the outside pressure to cause its expulsion by its own expansion during the exhaust stroke. Thus from boiler to condenser the steam is continuously expanding and undergoing liquefaction corresponding to the mass and molecular work of such expansion, and the heat transmuted into this work or annihilated, is so large a quantity that, as a practical average from good engines, only from 75 to 80 per centum of the heat imparted to the feed water in the boiler is recovered in the injection water of the condenser.

After the closing of the cut-off valve, the resulting expansion of the steam in the cylinder does mass and molecular work. The mass work is overcoming the aggregate opposed resistances, including in this term the friction of the moving parts

of the mechanism, the back pressure against the piston, the external or commercial load, and the friction due to that load, all for only the portion of the stroke of the piston performed by the expanding steam. The whole of this work done by the expanding steam is given in foot-pounds by the mean pressure of that steam on the piston, as shown by an indicator diagram in pounds per square inch, measured down to the line of no pressure, multiplied by the area of the piston in square inches, and by the length in feet passed over by the piston during the expansion. This is the only manner in which the work done by the expanding steam can be obtained. It cannot be had by means of any formula whatever, or by any method of reasoning, for the expansion curve is due to so many factors, most of them unknown, and all of them variable, that it can be obtained only experimentally—by direct measuring. The number of foot-pounds of work done by the expanding steam divided by the Fahrenheit mechanical equivalent of heat, gives the number of Fahrenheit units of heat taken from that steam and transmuted into the mass work done by its expansion.

But the expansion of steam, while either doing or not doing external work, involves molecular work, resulting from the fact that steam is not an ideal gas, but is what may be called a typical vapor, possessing strong molecular forces to be overcome during its expansion, while an ideal gas possesses none, and this molecular work being additional to any mass work that may be done during the expansion, requires correspondingly additional heat for its performance, which heat must be taken from the expanding steam itself. The *total* quantity of heat in unit of weight of steam before and after expansion, is expressed by the latent heat, and this heat diminishes as the temperature increases. Now an ideal gas has no latent heat because it is not liquefiable, its atoms have no interaction upon each other, are not held together by any molecular attractions, and do not, consequently, require any force to separate or remove them farther apart. The atoms composing an ideal gas are simply infinitesimal masses subject to only the mechanical (as opposed to chemical) laws of momentum and *vis viva*. All vapors, sup-

posing them to remain in contact with the liquids from which they are evolved, approach the gaseous state nearer and nearer the more and more they are heated, the increase of temperature being experienced by the liquids as well as by their vapors, and being accompanied by corresponding increase of pressure. Now this nearer and nearer approach to the gaseous state means that the attractive molecular forces become weaker and weaker, until finally a temperature is attained at which the latent heat vanishes, and above this temperature, which is called the critical point, no pressure, however great, can liquefy the gas. The effect of the molecular forces has been sensibly extinguished, and cannot be made to re-appear by any proximity of the molecules due to pressure alone. When this condition is reached, what was a vapor has become a gas. Steam has disappeared, and its materials now form another substance—gaseous protoxide of hydrogen—a substance having different mechanical, chemical and physical properties. All the laws and methods which were true of steam, become non-applicable to the substance into which it has been transformed by heat. The temperature of this inversion has never been ascertained for steam. The only experiment on the subject is a very incomplete and imperfect one made a great while ago by Cagniard de Latour, who, judging from deceptive optical appearances, placed it at from 750 to 800° Fahr.; my own calculations make it 1557° Fahr., about double of those figures. Of course not even a remote approximation to such temperatures can ever be employed in practice; but if they could be, the resulting substance would have the mechanical properties of other physical fixed gases, and could be cooled to any degree above the critical point without liquefaction, experiencing only reduction of temperature and corresponding lessening of bulk under constant pressure.

Below the critical point, however, steam is a vapor, and its action is governed by different laws. It is saturated at all temperatures when under corresponding pressures, a term denoting that its density is a maximum for its temperature, from which definition there results that its temperature cannot be reduced, not even infinitesimally, without a corresponding amount of liquefaction.

Now, saturated steam expanding in a cylinder and doing work is always in the saturated state. Its pressure and temperature lessen as the expansion increases, but it remains saturated throughout the whole of its expansion; nay, more, it is not *dry* saturated steam, but very *wet* saturated steam throughout, being more or less loaded with finely comminuted water in the state of haze or mist, and present from the causes previously described. Not the slightest refrigeration can be operated upon this steam without producing a corresponding liquefaction. And here is the direct issue between me and Mr. Allderdice. My method of calculating the weight of steam liquefied when working expansively is described in "A Lecture on the Processes of Steam in its Development of Power by Means of a Steam Engine," delivered before the Sibley College of the Cornell University, December 13, 1889, and printed in the October and November, 1890, issues of The Journal of the Franklin Institute, as follows:

"The quantity of heat or molecular *vis viva* transmuted by expanding steam into the external work developed by the expansion can be calculated. Accepting 789¼ foot-pounds of work as the mechanical equivalent of one Fahrenheit unit of heat, the division of the number of foot-pounds of work done by the expanding steam by 789¼ gives the number of Fahrenheit units transmuted into that work; and if this quotient be divided by the mean latent heat of the expanding steam the quotient will give the number of pounds weight of steam liquefied to furnish the quantity of heat."

The above method of calculation is both complete and correct, and will give exact results, the mean pressure of the expanding steam being always measured from the *indicator diagram* and down to the zero line. I may, however, add a few remarks on a germane matter, which has been overlooked by even such scientists as Rankine. The water of liquefaction, resulting from the molecular and mass work of the expanding steam, has, of course, the temperature of the steam from which it was precipitated, and as the continuously decreasing pressure of the steam, after the closing of the cut-off valve, is accompanied by a correspondingly continuous lowering of the boiling point, a portion of the above

water of liquefaction formed under the higher pressures is revaporized under the lower pressures, and the resulting steam mingling with the other steam present and undergoing expansion with it is, like that steam, liquefied in part to furnish the heat transmuted into the work of expansion. When the pressure of the expanding steam is obtained from an indicator diagram the effect of all this is included, and no especial allowance for it is needed; but if the pressure of the expanding steam be calculated from a formula, this effect is excluded, and here will be recognized one of the reasons which makes any such calculation impossible. The true curve of the expanding steam can only be traced graphically by the steam itself acting upon the spring of an indicator.

In making numerical calculations, I have always taken the latent heat of the mean pressure of the expanding steam instead of the mean latent heat of the expanding steam, because of the great labor required to ascertain the latter, and because the numbers given by the former are nearly the same as by the latter. The weight of steam in the cylinder is continuously varying during the expansion, owing to the continuous liquefactions and revaporizations due to all causes. Hence, in order to ascertain the mean latent heat of the expanding steam, the expansion portion of the stroke of the piston, as shown on an indicator diagram, must be divided into a number of equal parts—the greater the number the more accurate the result—and the mean pressure and the weight of steam ascertained for each part; then the product of the latent heats due to these pressures multiplied by the respective weights of steam, being determined for each part, the sum of these products divided by the sum of the steam weights will give the mean latent heat of the expanding steam, approximately, of course, but with a closer and closer approximation as the number of parts is greater and greater.

Mr. Allderdice declares that my method of ascertaining the weight of steam liquefied by expansive working as set forth in the above citation from the Sibley lecture is erroneous, and he says of it as follows: "The error of this method of computa-

tion lies in the assumption that the heat transformed into work during expansion is all given up by the steam that is condensed, and none of it by the steam that remains uncondensed. But as a matter of fact we know that the uncondensed steam remaining in the cylinder at the end of the expansion has given up a portion of its energy; its temperature and pressure have been lowered from what they were at the beginning of the expansion; its total heat per pound has been reduced; consequently, any method of computation of the amount of steam condensed during expansion which fails to take into account the loss of energy of the uncondensed steam must give a result that is too large."

The error in the above statement by Mr. Allderdice is fundamental. He says, substantially, that the abstraction of heat from steam which is saturated and remains saturated after the abstraction when working expansively, produces two effects; one, liquefaction, corresponding to a portion of the abstracted heat; the other, a reduction of the heat in the remaining steam due to the difference between its temperatures before and after expansion. That heat can be abstracted from steam which is saturated and remains saturated during and after the abstraction, without producing a liquefaction equivalent to the whole of the heat abstracted, is a physical impossibility as deducible from the very definition of saturated steam; and he makes no attempt to reconcile this opposing fact of nature with the imaginary fact of his assumption, nor to explain how the effects he asserts can possibly be physically produced. He has attributed to a very vaporous vapor the qualities that belong to only the permanent gases. He says, further, that the total heat per pound of the steam remaining in the cylinder at the end of the expansion has been reduced. And these latter words furnish the key to all his errors on this particular subject. He entirely mistakes what the total heat of steam is, and the result of that mistake has carried him so far from the truth that the exact reverse of his statement is true. What I mean to say is, that the total heat per pound of the steam remaining in the cylinder at the end of expansion has been increased instead of reduced, and increased in the direct ratio of the latent heats due to the pressures before and after ex-

pansion. The latent heat of steam is its total heat, and increases with lessening pressures. There is no other total heat of steam than this, for when the latent heat is abstracted the steam changes to water, an entirely different substance, with different physical, mechanical and chemical properties. That the water of liquefaction has the temperature of the steam liquefied is quite true, but neither this temperature nor the heat in this water is concerned in the present problem, which is purely the effect on saturated steam of the abstraction of heat from it, and ends with the transformation of the steam into water. Mr. Allderdice's total heat of steam is something very different, and nothing to the purpose. His total heat is the sum of the latent heat of the steam and of the heat in the water of liquefaction between the temperatures of liquefaction and the freezing point of water; but such a total heat has no more connection with the subject than the diameter of the moon. The total heat in the steam, and the aggregate heat in the steam and in the water from which the steam was evaporated, measured down to the arbitrarily chosen temperature of 32 degrees Fahrenheit as a base, are very different quantities, and have absolutely no connection. Any other temperature might just as well have been selected as a base, in which case the results of calculation made according to Mr. Allderdice's data would have been different—a manifest absurdity.

Regnault used the expression "total heat of steam" to denote the quantity of heat required to raise the temperature of water from the centigrade zero, or 32 degrees Fahrenheit, to the temperature of the boiling point under the given pressure, and then to vaporize the water under that constant pressure. He first ascertained the latent heat of the steam as previously described, and then, by an independent set of experiments, ascertained the specific heat of water at different temperatures, and as this specific heat gave the quantity of heat in the water at the respective boiling points, the addition of that quantity to the latent heat gave a sum to which he applied the term "total heat." Now, of the two components of this total heat, one, the heat in the water, increases with the pressure, while the other, the heat in the steam or the latent heat, decreases with the pressure, and as the former increases more than the latter

decreases, this kind of conventional total heat, as a whole, increases with the pressure. Mr. Allderdice's error consists in thus applying it, when, in fact, it has no application at all.

I have said that the pound of steam after its expansion in the cylinder contains more heat than before its expansion, an obvious fact from its greater latent heat; now this additional heat must be obtained from the expanding steam itself, as there is no other source, and if this additional heat be taken from the steam there must result a corresponding liquefaction of a portion of that steam, a liquefaction of so much of it as will liberate the additional heat needed. Thus the expansive working of steam entails a liquefaction of it sufficiently great to free, not only the heat required for the external work done by the expansion, but also for the internal work accompanying that expansion. The reason why the latent heat of steam of lower pressure is greater than the latent heat of steam of higher pressure, is that the molecular forces act more strongly at lower pressures, because the temperature is then lower, than at higher pressures, because the temperature is then higher. The *vis viva* of the molecules is greater at the higher than at the lower temperature, and the constant molecular forces produce proportionally less effect.

From the foregoing will be perceived that the heat required for transmutation into both the external and internal work done by expanding steam, is wholly obtained from the liquefaction of a sufficient portion of the steam to free it, and that none of this heat is from the expanding steam without undergoing liquefaction, the steam containing, in fact, more heat per pound after than before its expansion. Throughout the whole expansion the steam remains in the saturated state, always possessing just heat enough per pound for that purpose, and, consequently, it cannot be cooled—not even to an infinitesimal degree—without corresponding liquefaction, for then there would be the contradiction of a vapor having a greater density than the maximum due to its temperature and pressure, a manifest physical impossibility. Finally, the expanding steam, from the beginning to the end of the expansion, is intimately mingled with more or

less finely comminuted water having the temperature of the steam surrounding it, and undergoing changes with changes of that temperature.

Mr. Allderdice cites Clausius as an authority in support of his method, but the formula of Clausius is quite as faulty as his own, though in a very different way, and is so far from supporting him that it is based on essentially different assumptions. In fact, Clausius employs, as I do, the latent heats before and after expansion as expressing the total heats in the steam per pound. But he immediately falsifies this truth by modifying the latent heats proportionally to the absolute temperatures of the steam before and after its expansion. He has not perceived that the latent heats include in themselves the effect of the different temperatures above the absolute zero of heat. Furthermore, the formula of Clausius attempts to solve the problem without the aid of an indispensable factor, namely, the number of foot-pounds of work developed by the expanding steam. Now, without this quantity, the problem is insolvable; for how can the quantity of heat transmuted into work be known, unless the quantity of that work is first known. The work done by the expanding steam is not a function of the difference of its pressures before and after expansion, but depends wholly on the curve of pressures made by the expanding steam in its continuous passage from the higher to the lower pressure—a controlling fact which the formula of Clausius silently ignores.

Mr. Allderdice also cites Rankine as an authority in his support, but Rankine, in his original formula, does not agree with him, or with himself at different times, nor with the truth at any time. In article 283, page 385, of "A Manual of the Steam Engine and other Prime Movers," published in 1859, Rankine gives, to determine the liquefaction of steam working expansively, a formula for "the proportion borne by the condensed steam to the whole mass of steam and water." The formula is based on wholly wrong assumptions, and contains factors having no connection with the subject, though, like Clausius, he considers the latent heat of steam before and after expansion, modified by the difference of the corresponding absolute tem-

peratures, to represent the total heat; and, like Clausius also, he has no factor of the foot-pounds of work done by the expanding steam. This formula is as fallacious as that of Clausius, and there is no warrant in the physics of the subject for the assumptions involved.

Rankine, in his latest utterance on this subject (1868), gives in round numbers, under the head of "Liquefaction of a Vapor by Expansive Working in a Non-conducting Cylinder," the following numerical example for steam:

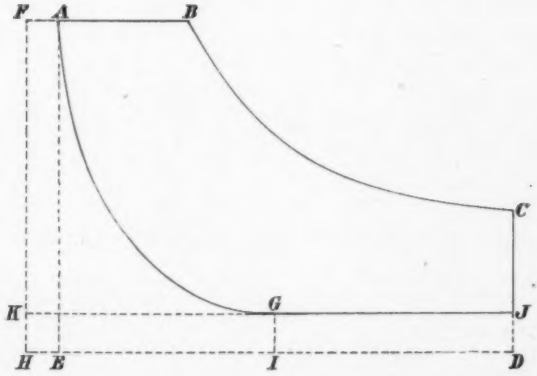
Pressures in pounds on the square inch:	
At beginning of expansion, absolute	115.
At end of expansion, absolute	29.
Rate of expansion, about	3.67
Work done by the expansion per pound of steam, about	84,000 foot-pounds.
Mechanical equivalent of a unit of evaporation at pressures between the above limits, about	730,000 foot-pounds.
Proportion of steam which would be liquefied if the total heat of evaporation were constant, 84,000 ÷ 730,000 =	0.115 nearly.
Subtract proportion of spray which would be dried up by wiredrawing with the same limits of pressure: fall of temperature, 90°F. ÷ 3,000 =	0.030
Proportion of steam liquefied in a non-conducting cylinder	0.085

The method shown in the above example, so far from supporting, as an authority, Mr. Allderdice's contention, is, on the contrary, exactly the same as my own with the exception that Rankine has subtracted 0.030 for steam due to the revaporization of part of the water of liquefaction resulting from the expansion of the steam *per se*, or "wiredrawing," as he terms it, quite overlooking the fact that this steam mingles with the mass of other expanding steam in the cylinder, does work there, and is partially liquefied in turn, as explained in the earlier part of this paper. Evidently, therefore, no deduction of 0.030, or any other quantity, should be made.

Rankine, in the above example, *assumes* for the work done by the expansion of one pound weight of steam from the pressure of 115 pounds per square inch above zero, to the pressure of 29 pounds per square inch above zero, 84,000 foot-pounds roundly. Then he takes for the product of the mechanical equivalent of heat and the latent heat of one pound of steam of

the mean pressure given by the expansion between the above limits of pressure, 730,000 foot-pounds roundly, dividing the 84,000, by which he obtains 11½ per centum of the total weight of steam undergoing expansion, as the fraction liquefied by the work of that expansion: a correct result, and the same that would be produced by my method, which Mr. Allderdice says is erroneous and contrary to Rankine's. As the 84,000 foot-pounds of work done by the expanding steam includes the work done by the steam resulting from what Rankine calls "wiredrawing," and which also does work expansively included in the 84,000 foot-pounds, there is no reason for making the dubious calculation of 0.030, nor for using it after it is made.

Mr. Allderdice next objects to my method of determining the total horses-power developed in the cylinder by the steam drawn from the boiler, when there is any "cushioning" or compression of the back pressure in the cylinder.



The total horses-power are understood to represent the entire dynamical effect of the steam, including overcoming the back pressure against the piston; and "cushioning" is understood to be the compression of that back pressure by the piston after the exhaust port is closed previous to the piston reaching the end of its stroke. To assist the conception, let the accompanying figure represent an indicator diagram, the closing of the exhaust

port taking place at such a point, *G*, that the back pressure *GI* is compressed to exactly the initial pressure *AE* of the steam drawn from the boiler. The area *FAEH* represents the steam space at the end of the cylinder in clearance and steam passages. Now whether there be cushioning or not, the area *ABCDE* represents the total work done per stroke of piston, and the line *FB* represents the total steam which does that work. If there be no cushioning, the steam *FB* will be wholly drawn from the boiler with the exception of the quantity of back pressure steam in the clearance and steam passage represented by *KHE;* but if there be the cushioning described, the steam *FB* present in the cylinder will not be wholly drawn from the boiler, because the portion of it represented by *FA* will be produced in the cylinder by the compression of the back pressure. The problem, now, is to ascertain what portion of the area *ABCDE* is produced by the steam *AB* drawn from the boiler. As the exhaust port closed at *G*, imprisoning in the cylinder the back pressure steam contained between *G* and the valve of the cylinder, and as this space contained at that time no other steam, evidently the area *GIEA* was formed entirely of the compressed steam, leaving the remaining area *ABCDIG* to be necessarily formed by the steam drawn from the boiler; the sum of these two areas being *ABCDE*, and the sum of the two steams being *FB*. I believe there is no escape from the rigor of this deduction. My method is founded on it, and may be stated as follows:

The total pressure on the piston, when the back pressure against it is cushioned, is the indicated pressure plus the mean back pressure against the piston above zero, exclusive of the cushioning, or for the length of the stroke of the piston between the commencement of the return stroke and the point in that stroke at which the cushioning commences, reduced in the ratio of that length to the whole length of the stroke.

The indicated pressure is represented by the mean height of the area *ABCJG*. Now, there is absolute certainty that this area is produced by only the steam drawn from the boiler, and as the whole of the steam drawn from the boiler must be exhausted per

stroke of the piston, and as the exhaust continues until the piston reaches the point *G*, there inevitably results that not only the indicated pressure, but the back pressure between *J* and *G* exhausted with it, is due wholly to the steam drawn from the boiler.

The small difference in the results computed according to the method which Mr. Allderdice recommends, and according to my method, he attributes on my part to an alleged "failure to eliminate entirely the action of the cushion steam in order to study the action of the working steam." There will be seen, however, from what precedes that, by showing the compression portion of the indicator diagram taken down to zero to be composed wholly of the compressed steam, and by eliminating this portion from the total diagram taken down to zero, leaving the remainder of the diagram to be composed wholly of the working steam, I have entirely eliminated every mechanical effect assignable to the compressed steam.

Further, Mr. Allderdice, after stating the difference he finds according to his view of the case and according to my method in the areas of the diagrams representing the total horse-power developed per stroke of piston by the working steam, says: "This small area represents the total horse-power per stroke developed by the cushion steam in expanding from the final forward pressure to the back pressure, during which expansion it performs work in pushing out the exhausting steam against the existing pressure of that steam." The fallacy in this statement consists in its assumption that the compressed steam does more work than is put into it by the compression—a physical impossibility. The mass work done by the compressed steam is simply the work of its own compression, and all other work done in the cylinder, including the expulsion of the exhaust, is done by the steam drawn from the boiler. From a dynamical point of view there is neither economic loss or gain by cushioning. If Mr. Allderdice's assumptions were correct, a distinct creation of force —force obtained from nothing—would result from the employment of cushioning.

I now come to Mr. Allderdice's arguments against my method

of determining the total horses-power developed by each cylinder of a compound engine. That method is as follows:

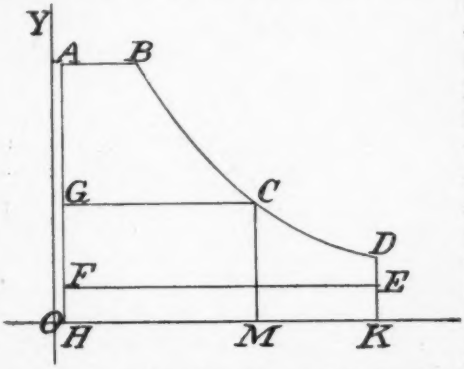
The indicated and net horses-power developed in the small and large cylinders are calculated for the respective indicated and net pressures on the *entire* areas of the pistons of those cylinders. The total horses-power developed in the small cylinder is calculated for the *entire* area of the piston of that cylinder, and for the total pressure measured down to the zero line. The total horses-power developed in the large cylinder are calculated for the total pressure therein measured down to the zero line, but only for the area of the "*annular ring*" remaining after the subtraction of the area of the piston of the small cylinder from the area of the piston of the large cylinder, because the back pressure against the portion of the piston of the large cylinder equal to the area of the piston of the small cylinder, is included in the back pressure against the piston of the small cylinder. The total pressure on the "annular ring," just described, is the indicated pressure on the piston of the large cylinder plus the back pressure against it.

This method is, of course, for compound engines whose cylinders have the same stroke of piston, and whose pistons make stroke for stroke. If the pistons of the two cylinders have different lengths of stroke, then the capacities of the cylinders must be used instead of the areas of their pistons; the "annular ring," with equal strokes of pistons, expressing the difference between the capacities of the two cylinders. The numerical results are exactly the same in both cases, and the principle is the same.

Formulas and graphic representations are only misleading in physical science unless based upon correctly ascertained processes of nature. Their whole value depends upon their correctly embodying those processes, and they stand or fall as they are in accord with or depart from this principle. A failure to comprehend and to be guided by this fact has rendered a vast mass of mathematico-physico disquisitions and abstract investigations only of use to establish a difference between what is called practice and theory, but which shouid be called practice and erroneous theory; for between the teaching by true theory

and by practice, which latter is only the teaching by infallible nature, there can be no difference.

The immediately above paragraph applies exactly to Mr. Allderdice's argument against my method of determining the distribution of the total horses-power developed by compound and triple-expansion engines. That argument is based on a complete misconception of the structure of such engines and of the processes of the steam in its development of power through their means, and he has egregiously misinterpreted the figure he has given in illustration—a figure which does not represent what occurs and what always must occur in nature. The figure in question, reproduced in the margin, supposes that, in a compound engine, there exists simultaneously on the piston of the large cylinder a different pressure on a portion of that piston (an area equal to the area of the piston of the small cylinder) from what exists on the remaining portion (the "annular ring"), manifestly a physical impossibility. In the figure referred to the line *HM* represents the volume of the small cylinder, and the line *HK* represents the volume of the large cylinder, the difference *MK* of these lines representing, of course, the difference of the two volumes, or the volume due to the "annular ring" formed by subtracting the area of the piston of the small cylinder from the area of the piston of the large cylinder. Now it is absolutely impossible for any difference of pressure to exist on the above different portions of the piston of the large cylinder, which portions are separated by only an imaginary line assumed for the purposes of calculation. Whatever pressure is upon the "annular ring" portion, the same pressure must inevitably be on



the portion within that ring—that is to say, the pressure upon each unit of the area *GCDKH*, representing the area of the piston of the large cylinder, must be exactly the same; nevertheless, Mr. Allderdice has not only entirely overlooked this self-evident fact, but has actually assumed different pressures to simultaneously exist on these two portions of the same piston, namely, he assumes for the "annular ring" portion the pressure represented by the mean ordinate of the area *CDKM*, and for the portion within the ring he assumes the quite different and much larger pressure represented by the vertical line *CM* or *GH*, and on this wholly unwarranted assumption of an impossible phenomenon he bases his argument against my method, the phenomenon being that the same elastic fluid has different pressures in the same space at the same time. The deductions drawn from such a misconception can only be absurdities, false to nature and inconsistent with themselves.

Returning now to Mr. Allderdice's diagram, which he has constructed on the following suppositions, namely: He supposes a compound engine, and, for simplicity of treatment, he supposes that there is no "cushioning" in either cylinder, no waste steam in clearance at the ends of either cylinder, that the steam is expanded in the small cylinder, and that in the large cylinder the cut-off valve closes when the piston of that cylinder has displaced a bulk equal to the bulk of the small cylinder, so that there is no "drop" or fall of pressure between the cylinders.

Mr. Allderdice now proceeds: "The combined card of such an engine would be represented by *ABDEF*." This assumption is inadmissible; there is no possibility of combining the cards taken from the small and large cylinders. A great variety of such combinations have been suggested, but none of them represent the phenomena, and none ever will, because the effect or true action of the "annular ring" in question cannot be made to appear in any combination, the combined diagrams having always to be supplemented by purely mental perceptions which cannot be graphically expressed. I accept Mr. Allderdice's diagram, which is as far from the truth as the rest, as only an arbitrary combination for giving precision to his argument. He continues:

"The total power per stroke for the whole engine would be represented by the area *HABDK*." This assumption is correct, as is also the next, namely, that "the total horse-power per stroke for the small cylinder, is represented by the area *HABCM*," but he immediately follows it with the incorrect assumption that the "total horse-power per stroke for the large cylinder would be represented by the area *MCDK*." The total horse-power developed in the large cylinder will be represented in this connection by the "annular ring" *MK* multiplied by the mean ordinate of the area *HGCDK*, which ordinate represents the total pressure on this "annular ring." With this correction made, Mr. Allderdice's figure proves the very fact he devised it to disprove, namely, that the total horses-power developed by a compound engine is the sum of the total horses-power developed in the small cylinder, using for the pressure upon the piston of that cylinder the mean pressure shown by the indicator measured down to the zero line, and, in the large cylinder, using for the pressure upon the piston of that cylinder the mean pressure shown by the indicator measured down to the zero line, and for the piston area the "annular ring" remaining of the piston of the large cylinder after deduction of the piston of the small cylinder. This is precisely my method when the strokes of the pistons of both cylinders have the same length, and the only one possible by which the total horses-power developed by each cylinder can be ascertained. And this method is always exact and just as applicable for "receiver" compounds as for "tandem compounds," or any other possible arrangement of compounds under any circumstances whatever. It is based upon the essential principle of the structure of all compound engines, and of all triple or more expansion engines. No intelligible theory of these engines can be given without it; neither can any complete calculations on the processes of the steam in them be made without it; nor, in fact, can these processes be even followed without it, as Mr. Allderdice will quickly find whenever he attempts to reduce his abstract speculations to concrete numbers—whenever he substitutes for arbitrarily ar-

ranged diagrams, which he fails in interpreting, cases of actual practice involving numerical calculation.

Finally, Mr. Allderdice says that: "The mean ordinate of the area *HGCDK* represents the indicated pressure, plus the back pressure for the large cylinder;" and immediately preceding this he said that "the total horse-power per stroke for the small cylinder was represented by the area *HABCM*." Now, as this latter area includes the area *HGCM*, which is also included in the former area, there follows that he has twice reckoned the area *HGCM* in the total horses-power developed by the engine. He has included it once in the total horses-power developed by the small cylinder, and again in the total horses-power developed by the large cylinder—a most obvious error.

In a compound engine, under the conditions imagined by Mr. Allderdice, the back pressure against the piston of the small cylinder is the same as the steam pressure on the piston of the large cylinder between the commencement of its stroke and the point of cutting off the steam, in which case the total horses-power developed by the engine, as a unit, will be the sum of the indicated horses-power developed in the two cylinders plus the horses-power required to overcome the back pressure against the piston of the large cylinder for its whole area, and the result will be evidently the same whether calculated according to this simple method or according to my "annular ring" method, the latter having the advantage, however, of giving the total horses-power developed separately by the small and large cylinders, instead of the aggregate only. But in a practical compound engine there is always a "drop" or loss of pressure shown on the indicator diagram, between the back pressure against the piston of the small cylinder and the steam pressure upon the piston of the large cylinder, counting the latter pressure from the commencement of the stroke of the piston of the large cylinder to the point of cutting off. This loss of pressure or "drop" is the inevitable result of transferring the steam from one cylinder to the other, and it may be little or much according to mechanical conditions; but it is always an economic loss, and should be made as small as possible. Now, this "drop" or loss

of pressure between the two cylinders may be considered, for the purposes of calculation, as the back pressure against the piston of the small cylinder resulting from imperfect action of the steam in passing between the two cylinders, just the same as the imperfect action of the exhaust steam in passing from the large cylinder to the condenser produces the back pressure against the piston of that cylinder. Hence, if the horses-power due to the "drop" or difference of pressure above described, and to the area of the piston of the small cylinder, be computed and added to the sum of the indicated horses-power developed in the two cylinders plus the horses-power required to overcome the back pressure against the piston of the large cylinder for its entire area, the result will be the total horses-power developed by the engine as a unit, and will be precisely the same as though calculated by my "annular ring" method.

The latter method, however, has the advantage of easier calculation, and of greater accuracy of data, as the exact determination from the indicator diagrams taken from the two cylinders, of the mean pressure representing the "drop," is both troublesome and uncertain. Although the total horses-power developed by the engine as a unit, can be calculated by the method above described just as well as by my "annular ring" method, the two results being numerically identical, yet only the latter method can be used when the liquefaction of steam due to its expansive working is to be calculated, for then there must be known the mean latent heat of the expanding steam producing the total horses-power developed by the expansion alone in each cylinder. Hence, this power must be obtained separately for each cylinder, and it cannot be had by any other than the "annular ring" method, which indeed is the proper and philosophical one to use in any case, as this method follows the natural processes of the steam in its development of power by means of the compound engine.

But Mr. Allderdice proceeds from bad to worse when he offers the following:

"Again, we may conceive of a compound engine with two cylinders of the same size, without cushion in the second cylin-

der, and with cushion at, say, three-fourths of the stroke in the first cylinder. In such an engine the volume of steam discharged from the first cylinder would be three-fourths of the cylinder volume, and the same steam when discharged by the second cylinder would have been expanded one and one-third times—that is, to the full volume of the cylinder; consequently the total horse-power developed by the steam during this expansion should be credited to the second cylinder. Yet by Chief Engineer Isherwood's method we should obtain no total horses-power from this cylinder, since no annular surface would remain after subtracting the area of the first piston from that of the second, the two pistons being of the same size."

The above statement means that a componnd engine can be made of two cylinders having the same diameter and stroke of piston, an utterly impossible thing with any arrangements of pressures, cut-offs or cushioning that can be conceived. Mr. Allderdice believes such an engine would work, but as he has imagined it, it never could make a single stroke of piston. He does not explain how, when one-fourth of the cylinder at one end is filled with steam compressed by one side of the piston to the boiler pressure, the steam of equal pressure on the opposite side of the piston, is to force that piston to the end of its stroke.

But even if this impossible effect were made possible, the argument would remain just as fallacious as before. Supposing that the engine could work—that is to say—supposing the pistons of both cylinders could make their strokes, in which case the space displacements of the pistons of the two cylinders would evidently be equal, the whole of the total horses-power developed by the engine would be developed in the first cylinder, as would also be the indicated and net horses-power, the second cylinder serving as only a portion of the exhaust pipe. Mr. Allderdice entirely overlooked the fact that the "annular ring" is only a measure of the difference of the space displacements of pistons of the same stroke and making stroke for stroke in the same compound engine. When he cushioned the steam in his first cylinder to the boiler pressure at three-fourths of the stroke of the piston, he doubtless intended that this piston should perform only

three-fourths of its stroke while the piston of the second or non-cushioned cylinder should perform the whole of its stroke, thus making the space displacement of the piston of the second cylinder one-third greater than that of the first cylinder. In that case he would have had a compound engine composed of two cylinders of equal diameters but unequal strokes of pistons, the difference of whose space displacements per stroke would have exactly represented the "annular ring" used in the case of two cylinders of equal strokes of piston and unequal diameters.

Lastly, Mr. Allderdice objects to my method of distributing the horses-power developed by an engine rotating the propelling screw of a vessel. One of the purposes, but not the only one, of that distribution, is to obtain the thrust of the screw, and I infer that Mr. Allderdice considers this thrust cannot be obtained from my distribution of the power because the portion of that power expended in the slip of the screw is calculated from the *apparent* slip which he thinks is different from the *real* slip. What is meant by the slip of a propelling screw? Simply the difference between the product of its pitch into the number of its revolutions during a given time, and the distance passed over by the propelled vessel in the same time. Did the threads or blades of the screw revolve in a fixed nut these two quantities would always be equal, and, if water could be conceived as acting with the immobility of a fixed nut, an infinitesimal area of blade surface would suffice for the propulsion without slip of the largest possible vessel at the highest possible speed, supposing water to be what it practically is, incompressible. But water is a very mobile substance, which resists, practically, only by its inertia, and yields to the slightest pressure if time enough be allowed, so that the acting blade surface, no matter how great in proportion to the resistance it overcomes, must experience a recession or slip as the screw revolves in the water. This recession or slip may be much or little according to the ratio of the acting blade surface to the thrust it delivers against the water, but whenever a screw revolves in water against a resistance there must be some slip, and, consequently, some power must be expended in producing it; that is, in giving backward movement to the water forming the reacting

mass to the surface of the blades. The same pressure is exerted equally upon the advancing vessel and upon the receding watery fulcrum, the two movements taking place simultaneously in opposite directions. The power developed, however, by this equal pressure will not be equal in both directions except in the single case where the two movements have equal speed, which only happens when the slip of the screw is 50 per centum of its speed. In all other cases the power expended in overcoming the resistance of the vessel will be to the power expended in the slip of the screw as the speed of the former is to the speed of the latter. But does the ratio of these powers, whatever it may be, affect the thrust of the screw, supposing the vessel to have always a constant speed? Evidently not, and for the following reasons:

Supposing the distance passed through the water in a given time by a vessel to be exactly ascertained experimentally, and supposing the pitch of the screw to be known exactly, also supposing the number of revolutions made in the same time by the screw to be known exactly, then the difference between the speed of the vessel and the speed of the screw will be the true slip, from which, under all circumstances, the power expended in that slip relatively to the power expended in the propulsion of the vessel, can be exactly calculated, let the physical phenomena of slip—of the actions of the screw on the water, or of the movements of water currents at the stern of the vessel—be what they may. There is no question of *real* or *apparent* slip in the matter, the problem is simply one of applied mechanics in the most elementary form. The screw is secured rigidly upon one end of its shaft, the other end of which presses with a given pressure against the vessel, giving movement to the latter, while, at the same time, the form of the screw recedes in the opposite direction—no matter from what cause or how: the thrust of the screw against the vessel forward and against the water backward must be the same from the equality of action and reaction: can there be then a possibility of a doubt that the loss of useful effect will be measured exactly by this recession? Mr. Allderdice thinks there is such a doubt, but I am sure he will not after he reads these lines. How, then, could he have formed

so baseless an opinion? Why, by the obfuscation produced by mixing up the physical phenomena of slip with my purely abstract method of calculation, whereas these phenomena, be they what they may, have no connection with it.

According to the principles above enunciated there can never be such a phenomenon as "negative slip." This term means that a body kept in motion by the immediate application of a point pushing in the direction of the motion of the body, has a greater velocity than the velocity of the pushing point, which is not only an utter mechanical absurdity in conception but a physical impossibility in fact. Even in the case of a vessel driven by the simultaneously applied pressure of sails and a screw, the latter cannot have a negative slip if it propels at all. It may, however, easily revolve in this case without propelling. It may either thread its way through the water with exactly the speed of the vessel, under which conditions it will, of course, have no slip, and the indicated power exerted by the engine will only be what is required to work it, *per se*, to overcome the friction produced in the machinery by the cohesive resistance of the water to the helicoidal surfaces of both sides of the blades in the direction of their helices, and to overcome that resistance itself. Or, the screw may have a less speed than the vessel, in which case the engine will be assisted by the sails, and the screw will be dragged bodily through the water by them, turning partly by the reaction of the water on the front surfaces of the blades and partly by the engine. Under these circumstances, and these only, will the screw show negative slip.

I am quite aware that screw vessels, under combined sails and steam, have shown a large power developed by the engine while the screw showed no slip, its speed being just equal to the vessel's speed, and Mr. Allderdice cites a case of that kind in a published report made by me of the performance of the U. S. S. *Trenton*. The apparent contradiction is easily reconciled. It is due to error in logging the speed of the vessel, which speed was unconsciously given too high by the deck officers, though accurately according to the log line. When a vessel is under sails with the wind acting upon them in any direction except dead

aft, she makes leeway; that is to say, she has two speeds, one in the direction of her keel, the other (the leeway) at right angles to it, the center of gravity of the vessel moving on the diagonal of the rectangle representing these two speeds. Of course, the diagonal is greater than either side of the rectangle, and as the log line necessarily gives the vessel's speed in the direction of the diagonal and not in that of the side representing her speed through the water in the direction of the keel, which is her true speed, the speed as logged is always too great by the difference. The speed of the screw, however, is unaffected by the leeway; it is always its speed in the direction of its axis, which is that of the vessel's keel. If this error of too much speed for the vessel be corrected in accordance with the above principles, the screw will always be found to have positive slip whenever the engine exerts more indicated power than is necessary to work itself, *per se*, to overcome the friction on the machinery due to the cohesive resistance of the water to the blade surfaces in the direction of their helices, and to overcome that resistance itself. I believe these causes are now pointed out for the first time, and their application will remove many anomalies.

In all the many investigations of the performances of screw vessels which I have had to make, I have always scrupulously given the data—the observed quantities—just as they were taken. I have never considered myself justified in introducing the slightest corrections.

In experiments where, from their very nature, all the data cannot be accurately obtained, as in the case of the lee-way above instanced, and in which tested instruments are not used, and in which the instruments that are used are probably in the hands of careless or incompetent persons performing their duties in a perfunctory manner, the results must always be taken as more or less approximative—not as exact. Statements must not be unhesitatingly received simply because authoritatively made, they must be critically examined, both as to the how and the who, and either accepted or rejected, or held as doubtful or in-

complete, according to their agreement with or departure from the indisputable principles of mechanical philosophy.

The thrust of the screw, however, can be calculated without either its slip or the speed of the vessel as factors, the steam pistons acting directly on the screw shaft. The effective area of the steam pistons, the length of their stroke, the number of double strokes made per unit of time, the pitch of the screw, and the remainder of the indicated pressure on the pistons after deduction of what may be grouped as the frictions and the resistance of the water to the screw blades, are the only data required.

Let A = effective area of pistons in square inches.

Let $2S$ = double stroke of pistons in feet.

Let P = pitch of screw in feet.

Let R = ratio of pitch of screw to double stroke of piston; that is $= \frac{P}{2S}$.

Let N = number of revolutions made by the screw per minute.

Let I = what remains of the indicated pressure on the pistons in pounds per square inch after deduction of the pressures equilibrating the two frictions, and equilibrating the cohesive resistance of the water to the screw blades in the direction of their helices at the velocity due to N.

Let T = the thrust of the screw in pounds.

$$\text{Then } T = \frac{A \times I}{R}.$$

Were it not that the cohesive resistance of the water to the screw blades is a function of the quantity N, varying in the ratio of the square of N, the quantity N could be eliminated; for of the friction pressures the one required to work the engine unloaded, or *per se*, is constant at all revolutions, and the other, required to overcome the friction of the load, is directly as the load.

The friction pressures, and the pressure expended in overcoming the cohesion of the water to the screw blades, can be calculated from experimentally known data with a sufficiently close

approximation for practical purposes. Of course, the above equation is exact.

The results obtained by employing the equation will be precisely the same as by employing the method I generally use and to which Mr. Allderdice objects, because he thinks the real slip of the screw is different from its apparent slip, evidently not knowing the mechanical reason for using slip in the calculation. I employ the latter method because it requires less time and labor. It is just as exact, but needs the speed of the vessel, and the difference between that speed and the speed of the form of the screw to be known as additional factors.

If the additional data be exact—that is to say, if the speed of the vessel be correctly given, the speed of the form of the screw being known from the product of its pitch into the number of its revolutions made per unit of time—the thrust of the screw calculated by both methods will be the same and has no connection with either the real or apparent (the apparent is the real if the data be true) slip considered physically, all that is needed being the difference between the speed of the form of the screw and the speed of the vessel. If the speed of the vessel be erroneously given, the thrust of the screw calculated by either method will be false; it will not be the thrust due to the true speed, but the fault will lie in the experimental data, and not in the method of using them.

There only remains to add that in all calculations of slip the *final* pitch of the screw is to be used when the latter has a pitch expanding from the front to the back edge of its blades. The common, but erroneous, practice is to use the *mean* pitch.

On page 35, 8th line, should be inserted after "cut-off valve," "and as occurs in the cylinder from the commencement of the stroke of its piston to the closing of the cut-off valve," &c.

V.

REPLY TO CHIEF ENGINEER ISHERWOOD, U. S. NAVY.

By Assistant Engineer W. H. Allderdice, U. S. Navy.

It cannot be denied that Mr. Isherwood has most vigorously defended his position. I regard his arguments as unsound, but they are so ably presented that I trust I may be pardoned for encroaching at some length upon the space of the Journal in answer. For clearness, I will divide the discussion as before into four sections, the first of which I consider the most important as involving fundamental principles to a greater degree than the others. I shall, therefore, devote to this first section the greater portion of my reply.

I. The Condensation of Working Steam during Expansion.

Mr. Isherwood begins by criticism of my statement that "the total condensation of steam in the cylinder of an engine using steam expansively may be considered as composed of two parts," &c., and endeavors to show that steam leaving the boiler as dry steam will be wet steam when delivered to the cylinder. Without inquiring at once into the truth of this, I will say that my article was concerned in this connection only with the condensation due to the work performed during expansion. I endeavored to show the error of Mr. Isherwood's method of computing this amount of condensation, holding the result given by his method to be too large. I *assumed* the steam at the beginning of expansion to be dry saturated steam, because there was nothing in his previous article to show that he treated it otherwise. I will endeavor to show, later on, that for wet steam the amount

of condensation during expansion will be less than for dry steam, so that if Mr. Isherwood's method is wrong for dry steam at the beginning of expansion, it is still farther in error for wet steam.

As to Mr. Isherwood's reasons for asserting that dry steam leaving the boiler will become wet steam by the time it arrives at the cylinder, although the question is one apart from the main point under discussion, it may be well to inquire into it, because it involves what appears to me to be a fundamental error in his system of thermodynamics. He says: "There can be no internal or molecular work done by saturated steam that is not accompanied and evidenced by this molecular liquefaction, and internal or molecular work can be done by steam only when it is expanded, in which case molecular work is always done by it whether or not the steam during its expansion does external or mass work." That is, Mr. Isherwood holds that expansion of saturated steam must be accompanied by liquefaction, whether external work be performed or not. Now it has been repeatedly shown, not only by theory, but by experiment, that saturated steam expanding without the performance of external work becomes, not partially liquefied, but superheated. It is scarcely necessary to cite authorities. Rankine, Clausius, Zeuner, Cotterill, Peabody and many others, may be quoted in support of this point. As far as my knowledge extends, Mr. Isherwood is the only writer who has disputed it. The observed fact of such superheating was one of the early arguments brought against the deductions of Rankine and Clausius relative to liquefaction during expansion. Rankine says ("Steam Engine and Other Prime Movers," § 283), after deducing the equation quoted in my article in the November issue of the JOURNAL:

"The principle just stated, as to the liquefaction of vapours by expansive working, was arrived at contemporaneously and independently by Professor Clausius and the author of this work in 1849. Its accuracy was subsequently called in question, chiefly on the ground of experiments which show that steam, after being expanded by being 'wire-drawn,' that is to say, by being allowed to escape through a narrow orifice, is superheated, or at a higher temperature than that of liquefaction at the reduced pressure. Soon

afterwards, however, Professor William Thompson proved that those experiments are not relevant against the conclusion in question, by showing the difference between the *free expansion* of an elastic fluid, in which all the energy due to the expansion is expended in agitating the particles of the fluid, and is reconverted into heat, and the expansion of the same fluid *under a pressure equal to its own elasticity*, when the energy developed is all communicated to external bodies, such for example, as the piston of an engine."

The eminent French engineer, M. Aug. Normand, in his article, "*Note sur la Machine à Vapeur*," in the December (1890) number of "*Mémoires et Comte Rendu de la Société des Ingénieurs Civils*," reviewing the economy trials of the French torpedo-boats, 126, 127, and 128, discusses at some length the superheating due to throttling when running at reduced speed, and ascribes in part to this superheating, with the attendant reduction of condensation due to the abstraction of heat by the metallic walls of the cylinder, the excellent economic results obtained.

The experiments of Professor C. H. Peabody, at the Massachusetts Institute of Technology (See "Flow of Steam through Orifices," and "Experimental Study of the Errors of Different Types of Calorimeters," in Proceedings of American Society of Mechanical Engineers, Vol. XI., 1890), may be adduced in evidence of the observed superheating of steam by free expansion. Prof. Peabody's throttling calorimeter is directly dependent upon this principle, the truth of which it demonstrates in every application. Additional proof, equally convincing if less scientific, lies in the fact, which I think most engineers have observed, that the bare hand may be held for some moments over a jet of escaping steam without being burnt; the steam is so dry, on account of the superheating due to wire-drawing, that it does not readily give up its heat; the first sensation produced, due to the absorption of the natural moisture of the hand by the dry steam, is rather cooling than otherwise, and it is only when the steam begins to condense upon the hand that any burning sensation is experienced.

The denial of the superheating of steam by free expansion,

and the assertion of the contrary effect, that is, of liquefaction, coupled with Mr. Isherwood's remarkable statement that "the latent heat of steam is its total heat," form a fundamental basis of error upon which his whole system of thermodynamics rests. I will not deny the logic of his deductions from these premises, but the premises appear to me to be incontestably false.

Confining attention for the present to the condition of the steam upon entering the cylinder, Mr. Isherwood's argument, as I understand it, may be briefly stated thus: there is an expansion in the steam pipe, evidenced by the difference of pressures between the two ends of the pipe, accompanied by mass work and molecular work, resulting in liquefaction; so that supposing dry steam to be supplied to the pipe at the boiler end, wet steam will be supplied to the cylinder. Let us consider this. That there is a gradual fall of pressure in the steam pipe from the boiler to the cylinder is undeniable; there is a loss of head, due to the frictional resistances of the pipe, the valves, and the ports; there is a further loss of head due to the change of velocity of the steam in passing from the pipe, where its velocity is high, to the cylinder, where it is low. The "mass work" performed by the steam, that is, the work of transportation from the boiler to the cylinder consists, except when steam is lifted against the force of gravity, of the work required to overcome the frictional resistances, and the work equivalent to the energy due to the velocity of the steam; the former is expended in the production of heat, whilst the latter is in great part re-transformed into heat when its velocity is reduced upon entering the cylinder. As to molecular work, it can not be denied that when the pressure of a saturated vapor is reduced, molecular work is performed, but the heat required to perform this molecular work is, for all vapors, less than the heat that would be given up by the reduction of the temperature of the vapor from that normal to the higher pressure to that normal to the lower; the consequence of which is that, unless the difference between these two amounts of heat is extracted by the performance of external work or otherwise, the vapor will be superheated at the lower pressure. It follows from this that when liquefaction occurs, it is due to the external work or "mass

work" performed, and in no sense to the molecular work, since the performance of molecular work alone would entail, not liquefaction, but superheat at reduced pressure. The conclusion which follows from these considerations is this, that since the "mass work" performed by steam in passing from the boiler to the cylinder is expended in overcoming frictional resistances, &c., resulting in the production of heat, the steam entering the cylinder, although somewhat reduced in pressure, possesses a temperature which, though lower than that normal to the boiler pressure, is higher than that normal to the reduced pressure at the cylinder; in other words, it is superheated. The effect has been that of "wire-drawing." It is assumed, of course, that the steam-pipe is non-conducting, but in practice we know that there is always a considerable loss of heat from conduction, no matter how thoroughly the steam-pipe may be lagged; this loss may, at times, offset the superheating of the steam occasioned by its "wire-drawing," and cause liquefaction, in which case wet steam will be delivered to the cylinder, but its wetness will not be due to the causes assigned in Mr. Isherwood's article.*

Mr. Isherwood is quite wrong in asserting that the transportation of the steam from the boiler to the cylinder has hitherto been supposed to be miraculously accomplished, without the expenditure of work. The subject may be found treated of at some length by Rankine in his "Steam Engine and Other Prime Movers," § 290, on the "Difference between Pressure in Boiler and Initial Pressure in Cylinder," and by Clausius in his "Mechanical Theory of Heat," Chapter XI., § 10, on "Changes on the Steam during its Passage from the Boiler into the Cylinder."

Passing over for the moment consideration of Mr. Isherwood's paragraph relating to the condition of the steam after admission to the cylinder and before cut-off, and coming to his treatment

* The extent of the loss of heat from conduction through lagged steam-pipes may be gathered from the report of experiments made by Prof. John M. Ordway for the Boston Manufacturers' Mutual Fire Insurance Company (an abstract of which is to be found in the London *Electrician*, January 16, 1891). For thirty-four "non-conducting" substances experimented upon, the average transmission of heat through a thickness of one inch, from steam at 310° F., was 210 British thermal units per square foot of surface per hour.

of the amount of liquefaction during expansive working, the main point upon which I took issue with him in my previous paper, I will quote one sentence which appears to me to contain the main source of the error of his method. He says: "That heat can be abstracted from steam which is saturated and remains saturated during and after the abstraction, without producing a liquefaction equivalent to the whole of the heat abstracted, is a physical impossibility, as deducible from the very definition of saturated steam." Now, saturated steam is defined as steam of maximum density at a given pressure, and at the temperature of evaporation corresponding to that pressure. If, by some means other than the performance of external work, the pressure be reduced, and simultaneously the temperature, from that corresponding to the higher pressure, to that corresponding to the lower, heat is abstracted without liquefaction; but if the reduction of pressure and temperature be accomplished by the performance of external work, the expenditure of heat required for the performance of that external work is, for steam at ordinary temperatures, greater than the amount of heat that may be abstracted without liquefaction, and consequently liquefaction ensues; but this liquefaction is not, as Mr. Isherwood says, "equivalent to the whole of the heat abstracted," but equivalent only to the difference between the whole of the heat abstracted and the amount that might be abstracted without liquefaction.

Mr. Isherwood's contention as to the amount of this liquefaction during expansive working, together with his assertion of the existence of an additional liquefaction due to the fall of pressure in the cylinder from the commencement of the stroke to the point of cut-off, may be dealt with together by a consideration of the full sequence of operations to which the steam is subjected, from its introduction in the boiler in the state of water to the completion of its expansion and partial liquefaction at the end of the stroke of the piston. Let us suppose the original temperature of the water to be that of the freezing point, not that this will in any way affect the results of our calculations, since we shall be concerned only with differences in amounts of heat, but simply because that is the temperature commonly as-

sumed as a base in calculated tables. The total amount of heat, H, to be communicated to one pound of water to raise its temperature from the freezing point to the temperature, t, and evaporate it at the latter temperature under the corresponding pressure, P pounds per square foot, or p pounds per square inch, may be considered as composed of two parts. Let us call the part required to raise the temperature of the water, q, and the part required to evaporate it, L. The latter is called the latent heat of evaporation, and may be further divided into the interior latent heat of evaporation and the exterior latent heat of evaporation. In explanation of these terms, I will quote from the paper on "Properties of Steam," by M. Dwelshauvers-Dery, published in the Proceedings of the American Society of Mechanical Engineers, Vol. XI., 1890:

"ρ, *interior latent heat of a pound of dry saturated steam*: that is to say, the heat which corresponds to the work necessary solely to overcome the internal molecular forces which hold the molecules at the distances at which they must be while the mass is in a liquid state, in order to bring them to the distances at which they will be found when the mass is in the condition of saturated steam.

"APu is the *exterior latent heat* of a pound of steam generated, under the constant pressure P pounds per square foot, while the piston sweeps through the volume, u. This demands explanation. The exterior latent heat has been measured under the following conditions: A pound of water is put in a cylindrical vessel at a temperature t, and in the cylinder is a piston which exerts on the surface of the water a pressure of P pounds per square foot, corresponding to t degrees for saturated steam. To give precision to the discussion, let $t = 292°.52$. The table gives $P = 8{,}640$ pounds per square foot. In this condition the pound of water occupies a volume, σ. Heat is furnished in a measured quantity. Then the piston rises in the cylinder. The temperature of the water remains unchanged, but the water evaporates little by little until there is nothing but dry and saturated steam at the pressure P, and the temperature t. At this instant, stopping the experiment, the volume u, traversed by the piston,

is measured and expressed in cubic feet. Hence follows naturally $\sigma + u = v$. Where v, as above, denotes the volume occupied by the pound of saturated steam at the pressure P, experiment shows that $P = 8{,}640$ pounds per square foot, and $t = 292°.52$.

"$\sigma = 0.0160$; $u = 7.0168$ cubic feet.

"Whence we conclude that the heat furnished has done a work Pu foot-pounds to raise the piston against the exterior pressure, and since each thermal unit corresponds to 772 foot-pounds, it follows that the heat which disappeared by doing this exterior work is denoted in thermal units by $\frac{Pu}{772}$, which becomes APu if $A = \frac{1}{772}$."

Considering the application of these principles to an engine, let us, for simplicity, imagine the cylinder to have its end opening directly into the steam space of the boiler. I think it is obvious that the piston of the engine replaces exactly the piston conceived of in the previous definition, the only difference being that, instead of pressing directly upon the evaporating water with a pressure of P pounds per square foot, it is separated by a cushion of steam of the same pressure. One pound of steam in evaporating displaces one pound of steam previously evaporated, forcing the latter into the cylinder, and moving the piston through the same space as if the newly evaporated pound had itself entered the cylinder directly. The work done is the same, APu. The pound of steam entering the cylinder has previously performed the same work in the same manner, so that we have in the cylinder one pound of steam to which an amount of heat, $H = q + \rho + APu$, has been communicated, and which has already performed in the cylinder an amount of work represented in thermal units by APu. The difference,

$$H - APu = q + \rho$$

represents the amount of energy above the freezing point still remaining stored in the steam and available for the performance of work by expansion. This is called the *intrinsic energy* of the steam above the freezing point. If, instead of one pound of

saturated steam, we have in the cylinder one pound of steam and water, a weight x being water and $1 - x$ steam, the heat which has been communicated to the mixture is represented by

$$xq + (1 - x) H,$$

and the work already performed, by

$$(1 - x) APu,$$

so that the *intrinsic energy* above the freezing point, for the mixture, is

$$xq + (1 - x) H - (1 - x) APu.$$

Now, I think it must be evident that if, during an adiabatic expansion, a pound of steam, or of mixed steam and water, passes from a higher condition as to intrinsic energy to a lower condition, the external work performed must be exactly equivalent to the loss of intrinsic energy. Applying this principle to the consideration of the case of one pound of dry saturated steam at a pressure P_1, possessing intrinsic energy above the freezing point equal to

$$H_1 - AP_1u_1,$$

which expands adiabatically whilst performing an amount of work represented in thermal units by W, a weight x being liquefied during the expansion, and the pressure being reduced to P_2, and the intrinsic energy above the freezing point to

$$xq_2 + (1 - x) H_2 - (1 - x) AP_2u_2,$$

we have

$$W = H_1 - AP_1u_1 - xq_2 - (1 - x) H_2 + (1 - x) AP_2u_2.$$

Substituting for q_2 its equivalent, $H_2 - L_2$ (L_2 being the total latent heat of evaporation at P_2, including both the interior and exterior latent heats), and solving for x, we obtain

$$x = \frac{W - (H_1 - H_2) + AP_1u_1 - AP_2u_2}{L_2 - AP_2u_2} \quad *$$

as the proportion of steam liquefied.

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* The expression for x may be more briefly stated

$$x = \frac{W - [(q_1 + \rho_1) - (q_2 + \rho_2)]}{\rho_2}.$$

This formula is due to Zeuner. The other form is adopted only for comparison with Mr. Isherwood's method.

To compare this with Mr. Isherwood's method, I will put it in the form,

$$x = \frac{W}{L_2} - \frac{H_1 - H_2}{L_2} + \frac{AP_1u_1 - (1 - x)\,AP_2u_2}{L_2}\ .$$

In my previous paper, in comparing Mr. Isherwood's method with an approximate method, the error of which I pointed out, I stated that, according to his method, the liquefaction during expansion would be indicated by $x = \frac{W}{L}$, in which L represents the mean latent heat during expansion. In the expression shown above, the third term is very small, much smaller than the second, whilst it is evident that the first term alone is smaller than $\frac{W}{L}$, so that the expression as a whole is considerably smaller than the result obtained by Mr. Isherwood's method.

The solution given above is full and complete, taking account of all transfers of heat in a non-conducting cylinder, including heat given up during the later stages of expansion by water condensed during the earlier stages, the mixed mass of water and steam being treated of as at the same temperature at the end of the expansion. The only drawback to its use is that it involves the work, W, performed during expansion. To take this from the indicator card of an actual engine is scarcely allowable, because the indicator card is affected by the transfer of heat to and from the metallic walls of the cylinder, involving condensations and re-evaporations which would not occur in a non-conducting cylinder. For this reason I think the formula of Rankine and Clausius preferable, since it involves only known data relative to the condition of the steam at the beginning and at the end of expansion.

I think it has been shown, in treating of the exterior latent heat of evaporation, that the work performed in a cylinder prior to cutting off the steam, is simply the equivalent of the exterior latent heat of the steam admitted; but Mr. Isherwood, arguing from the fact that there is a certain fall of pressure from the beginning of the stroke to the point of cut-off, asserts that this fall of pressure shows an expansion for which there must be a cor-

responding liquefaction. It is scarcely reasonable to suppose that, where there would be no liquefaction due to the work performed if the pressure were maintained constant and equivalent to the boiler pressure up to the point of cut-off, there will be liquefaction when this pressure falls off, and the work done before cut-off is reduced in amount. In actual engines, this fall of pressure is due principally to condensation caused by the abstraction of heat by the metallic cylinder walls. In a non-conducting cylinder this cause would not exist, and the only loss of pressure would be the loss of head due to wire-drawing, resulting, as in the case of loss of pressure in the steam pipe from the same cause, not in liquefaction, but in a superheating of the steam at the reduced pressure.

All the tendencies affecting the steam in its passage from the boiler to the cylinder, and in the cylinder up to the point of cut-off, barring conduction, are to make it drier; nevertheless, it may happen that the steam leaves the boiler in such a wet condition as to be still supersaturated at the point of cut-off. Applying the principle already used, that the work performed during expansion is equivalent to the loss of intrinsic energy, we have, calling the weight of water in one pound of mixed steam and water at the beginning of expansion, x_1, and at the end of expansion, x_2,

$$W = x_1 (H_1 - L_1) + (1 - x_1) H_1 - (1 - x_1) AP_1u_1$$
$$- x_2 (H_2 - L_2) - (1 - x_2) H_2 + (1 - x_2) AP_2u_2 \quad ,$$

from which the liquefaction,

$$x_2 - x_1 = \frac{W - (H_1 - H_2) + AP_1u_1 - AP_2u_2}{L_2 - AP_2u_2}$$
$$- \frac{x_1 (L_2 - L_1)}{L_2 - AP_2u_2} - \frac{x_1 (AP_1u_1 - AP_2u_2)}{L_2 - AP_2u_2} \quad .$$

The first term of this expression is the same as the whole liquefaction given by the formula for the case of dry saturated steam at beginning of expansion, from which it may be seen that the liquefaction produced by expansion is less in the case of wet steam at beginning of expansion than in the case of dry steam, so that Mr. Isherwood's method of calculation, which

gives results too large in the case of dry steam, is still farther in error for wet steam.

Mr. Isherwood does not believe in the solution of thermodynamical problems by means of mathematical analysis, but prefers rather the methods of philosophy; and I think a careful reading of his philosophical treatment of the subject of the liquefaction of steam during expansive working will justify the statement that the conclusions at which he arrives concerning the behavior of saturated steam should apply equally well to the case of any other saturated vapor. Yet it has been shown by experiment that some other saturated vapors behave, during expansive working at ordinary temperatures, quite differently from steam. To explain this it is necessary to introduce the definition of what the leading writers on thermodynamics term the Specific Heat of Saturated Vapor. Clausius says: "The specific heat here treated of is not, however, that at constant volume, nor yet that at constant pressure, but belongs to the case in which the pressure increases with the temperature in the same manner as the maximum expansive power of the saturated vapor." For one pound, it is "that quantity of heat which saturated vapor requires to heat it through 1°, if it is at the same time so powerfully compressed that even at the higher temperature it again returns to the saturated condition." For saturated steam at ordinary temperatures, this specific heat is negative; that is to say, if one pound of saturated steam, at a given temperature and corresponding pressure, be compressed until the pressure is increased to that corresponding to a temperature 1° higher, heat must be abstracted in order to maintain the steam in the saturated condition, otherwise it would become superheated. But this is not true for all vapors; with ether, the contrary is the case. The specific heat of saturated ether is positive, so that ether is liquefied by compression, but superheated by expansive working. The specific heat of saturated steam, negative at ordinary temperatures, approaches zero with increase of temperature, and theory indicates that at some high temperature its value becomes positive. But this "temperature of inversion," as it is called, lies beyond the practical range of experiment. The temperature of

inversion should not be confounded with the "critical point," or temperature at which a vapor acquires the characteristics of a fixed gas, referred to in Mr. Isherwood's paper as "the temperature of this inversion." It is a temperature considerably below the "critical point," and for some vapors, such as chloroform and bisulphide of carbon, the specific heats of which, like the specific heat of steam, are negative at ordinary temperatures, the point of inversion lies within the practical range of temperature and has been experimentally determined. An account of the experiments of Cazin and of Hirn is given by Clausius ("Mechanical Theory of Heat," Chapter VI.), but it will be sufficient here to recount the results. Observations were made by means of cylinders with glass sides. Temperatures were carefully noted. Liquefaction was evidenced in the experiments by the appearance of cloud, superheated and dry vapors being quite transparent. Steam formed a cloud during expansion, but remained quite clear during compression; ether remained clear during expansion, but formed a cloud during compression; for chloroform, the point of inversion was calculated by Cazin at 123°.48 C., and was experimentally determined at about 130° C. Up to 130° C., the vapor of chloroform formed a cloud during expansion, but remained transparent during compression. Above 136°, it remained transparent during expansion, but formed a cloud during compression. The slight discrepancy between the calculated and the experimental results was attributed by Cazin to the fact that the chloroform used was not chemically pure.

As to Mr. Isherwood's statement that "the latent heat of steam is its total heat," I will not attempt to argue; it appears to me to carry with it its own refutation; it bears us to the conclusion that we may abstract heat from a body, and have more remaining in it after the abstraction than before. In my previous article, I used the term "total heat" in its conventional acceptance; the freezing point of water is commonly taken as a base. Any other temperature may be selected equally as well, without involving (to quote Mr. Isherwood) "a manifest absurdity," since all the calculations into which "total heat" enters de-

pend only upon differences of total heat, not absolute amounts, so that the base is immaterial.

Mr. Isherwood is particularly hard upon Rankine and Clausius. I do not think they need any defense at my hands, but it may be well to refer to certain points regarding which their attitudes might be misunderstood from a reading of Mr. Isherwood's paper. He says of Clausius: "Clausius employs, as I do, the latent heats before and after expansion as expressing the total heats in the steam per pound. But he immediately falsifies this truth by modifying the latent heats proportionally to the absolute temperatures of the steam before and after its expansion"; and of Rankine: "Like Clausius, he considers the latent heat of steam before and after expansion, modified by the difference of the corresponding absolute temperatures, to represent the total heat." I must confess that I do not understand what this means. It does not appear to me to express a definite idea; but Rankine's definition of total heat is quite definite, reading as follows ("Steam Engine and Other Prime Movers," § 215):

"Total heat of evaporation, or *total heat of vapor*, is a conventional phrase used to denote the sum of the heat which disappears in evaporating one pound of a given substance at a given temperature (or *latent heat of evaporation*), and of the heat required to raise its temperature, before evaporation, from some fixed temperature up to the temperature of evaporation."

Clausius objects altogether to the use of the terms *total heat* and *latent heat*, and lays down ("Mechanical Theory of Heat," Chapter I., § 10) "the principle that all heat existing in a body is appreciable by the touch and by the thermometer"; and that the heat which disappears during fusion and vaporization "exists no longer as heat, but has been converted into work". For the term *latent heat*, he proposes the substitution of the term "*work-heat*."

Mr. Isherwood furthermore says of Clausius: "The formula of Clausius attempts to solve the problem without the aid of an indispensible factor, namely, the number of foot-pounds of work developed by the expanding steam"; and of Rankine: "Like Clausius, also, he has no factor of the foot-pounds of work done

by the expanding steam." To understand the methods of Clausius, one need only refer to "The Mechanical Theory of Heat," Chapter XI., § 12, "Determination of the Work Done during One Stroke." He expresses the work in four terms: The work done during the admission of the steam, the work done during the expansion, the negative work done during the return stroke of the piston, and the negative work done in forcing back the mass into the boiler. Clausius treats always of one pound of fluid, which may be partially in the condition of steam and partially in that of water; so, also, does Rankine in all his discussions of the behavior of steam in a non-conducting cylinder. In the formula for liquefaction during expansive working, given in different forms by both Rankine and Clausius, it is true that an expression for the work done during expansion does not enter, but the quantities introduced in the formula are themselves functions of the work done during expansion, so the result is the same as if an expression for that quantity had been itself introduced.

Mr. Isherwood leads us to infer, by the quotation of a numerical example from one of Rankine's later papers, that he did not hold, up to the time of his death, to the truth of the formula for liquefaction during expansive working given in his "Steam Engine and Other Prime Movers." The article from which the numerical example is taken may be found printed in the London *Engineer*, of February 21, 1868. A reading of the article will make it clear to anyone that the example was given only as a rough approximation; the reasoning being, that if we calculate the amount of liquefaction that would occur during expansive working if the total heat of evaporation were constant, and subtract from this the amount of water that could be evaporated by the superheat resulting from the same degree of expansion without the performance of work, the difference will represent roundly the actual liquefaction. If we take the formula previousiy given in this paper,

$$x = \frac{W}{L_2} - \frac{H_1 - H_2}{L_2} + \frac{AP_1u_1 - (1 - x)\,AP_2u_2}{L_2},$$

drop the last term, which is small, and substitute

$$H_1 = 1092 + 0.3\,(T_1 - 32°),$$

and $$H_2 = 1092 + 0.3\,(T_2 - 32°),$$

we obtain

$$x = \frac{W}{L_2} - \frac{0.3\,(T_1 - T_2)}{L_2},$$

application of which will give about the same process employed in Rankine's numerical example. So far, however, from forsaking his previous method in favor of the approximate calculation used in the numerical example quoted by Mr. Isherwood, Rankine refers in a foot-note to the "exact formula" given in his "Steam Engine and Other Prime Movers," and at the end of the article gives, in a "Mathematical Appendix," the derivation of this formula.

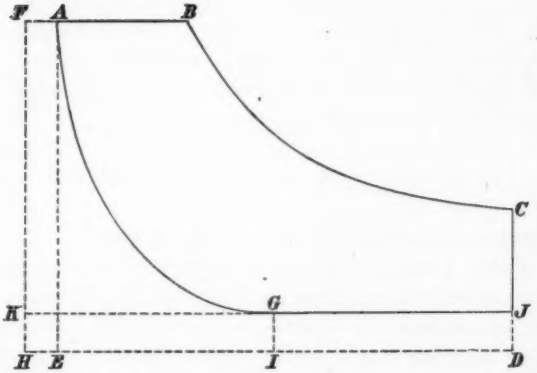
With regard to Mr. Isherwood's statement that Rankine's method, employed in the numerical example referred to, agrees with his own, "with the exception that he has subtracted 0.030," I can only say that this 0.030, or something very near it (amounting to about one-third of the total), is all that I have written about.

II. The Total Horse-Power of the Working Steam.

With regard to this point, the issue between Mr. Isherwood and myself may be stated very briefly. Reproducing Mr. Isherwood's diagram, I will quote from his article one sentence upon which his demonstration depends, and to the truth of which I can not agree. He says:

"Now whether there be cushioning or not, the area *ABCDE* represents the total work done per stroke of piston, and the line *FB* represents the total steam which does that work."

Upon this point we differ. I hold that the area *ABCDE* does not represent the total work done per stroke of piston; and for this reason: the area *ABCDE* represents the total work done by the steam in the cylinder when its pressure has been reduced to *DC*, but there is a further reduction of pressure from *DC* to *DJ*, during which a portion of the steam is blown out of the



cylinder, whilst that remaining in the cylinder performs work due to the expansion from the higher pressure to the lower, this work not being shown on the diagram. By the method given in my previous paper, the diagram is corrected to take account of this work; the action of the cushion steam is completely eliminated, and the corrected diagram represents simply the changes of volume and pressure undergone by the steam passing through the engine, or *working steam.*

The principle involved may be thus stated: Suppose we have a vessel filled with steam, and upon opening a cock, a portion of this steam is allowed to escape. Is work done by the steam in the vessel, or is it not? I hold, applying the principle laid down by Sir William Thompson, that in the case of the escaping steam we have *free expansion*, the tendency of which is to cause superheating, but in the case of the steam remaining in the vessel, we have *expansion under a pressure equal to its own elasticity*, consequently work is performed corresponding to that expansion. If this is not true, Mr. Isherwood's method of computing the total horse-power of the working steam is correct, and mine is wrong.

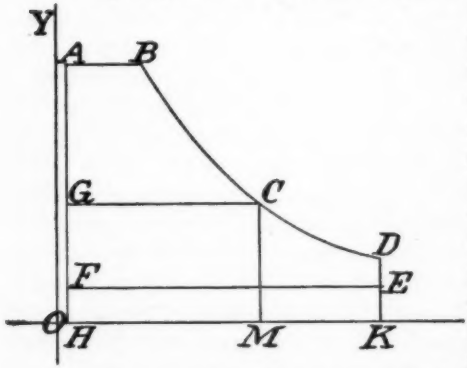
I think the error of Mr. Isherwood's method may be most clearly shown by carrying it to the extreme. Suppose the exhaust to open exactly at the end of the stroke, the piston to pause long enough to effect the reduction of pressure, and the

exhaust to be then closed, so that compression shall take place throughout the whole of the back stroke, then, according to his method, the total horse-power developed by the steam passing through the engine would be identical with the indicated power, a result the error of which is evident.

III. The Distribution of the Total Horse-Power.

Commenting upon my criticism of his "annular ring" method of accounting for the distribution of the total horse-power between the cylinders of a compound engine, Mr. Isherwood has given to my argument a construction very far from its intended meaning. Reproducing here the original figure, I will endeavor to make this clear.

For simplicity, and to separate the question from others upon which we were at issue, I made certain assumptions, namely: that, in an imaginary compound engine, there is no compression in either cylinder; that the final forward pressure in the small cylinder is the same as the back pressure in that cylinder; that clearance spaces are so small that their influence may be neglected; that there is an intermediate receiver between the cylinders so large that the pressure therein is practically constant; that steam is cut-off in the large cylinder at a volume equal to the total volume of the small cylinder; and that loss of pressure due to friction in steam passages may be neglected. With these assumptions, *HM* representing the volume of the small cylinder, and *HA* and *MC* respectively the initial and final pressures therein, *ABCG* represents the indicator diagram for



the small cylinder. Similarly, *HM* representing the volume of the large cylinder up to the point of cut-off, *HK* the total volume of that cylinder, and *HG* = *MC* and *KD* respectively the initial and final pressures therein, *GCDEF* represents the indicator diagram for the large cylinder. Referring these two diagrams to the same axes, we obtain a combined diagram, *ABDEF*, representing the series of changes of pressure and volume undergone by the steam in its passage through the engine. Mr. Isherwood is willing to allow that, upon such a diagram, the total horse-power developed in the small cylinder is represented by the area *HABCM*, and that the total horse-power for the whole engine is represented by the area *HABDK*. The unavoidable deduction is that the total horse-power developed in the large cylinder is represented by the difference between these two areas, or the area *MCDK*.

Now, according to Mr. Isherwood's method, the total horse-power developed in the large cylinder should be equal to the product of the length of stroke, into the area of the "annular ring" remaining of the piston of the large cylinder after subtracting from it the area of the piston of the small cylinder, into the mean pressure throughout the stroke on the piston of the large cylinder measured down to the line of no pressure; or, in other words, it should be equal to the difference of volume of the two cylinders multiplied by the mean pressure throughout the stroke on the piston of the large cylinder. If this be true, this product should be represented by the area *MCDK*, and my argument is contained in the proof that *MCDK* does not represent this product. The area *MCDK* is equal to the product of *MK* by the mean ordinate of *MCDK*. *MK* is equal to *HK* — *HM*, or is equal to the difference of volumes of the two cylinders; consequently if Mr. Isherwood's method be correct, the mean ordinate of *MCDK* should represent the mean pressure throughout the stroke in the large cylinder. But, obviously, it does not; from which it follows that his method is incorrect.

I cannot see anything in this demonstration to justify Mr. Isherwood's statement that I assume different pressures to simultaneously exist on two portions of the same piston. It might as

well be asserted that, because I say *HM* represents the volume of the small cylinder, I assume the same pressure to exist on the pistons of both the small and large cylinders.

I come now to the consideration of the second imaginary engine referred to in my previous article, a compound engine with two cylinders of the same diameter and of the same stroke of piston, with compression at three-fourths of the back stroke in the first cylinder, and without compression in the second cylinder. Mr. Isherwood says this engine would not run at all, but I fail to see why not. The back pressure in the first cylinder would simply be equal to the final pressure in that cylinder; the cushioning would not exceed what is frequently seen in practice; the initial pressure in the second cylinder would be greater than the back pressure in that cylinder. I do not suppose anyone would build such an engine, but it is quite certain that it would run if anyone should build it. Moreover, in such an engine, a portion of the total horse-power would be developed in the second cylinder, and for this reason:—since compression would occur at three-fourths of the back-stroke in the first cylinder, and not at all in the second cylinder, the volume of steam delivered by the first cylinder to the second would be three-fourths of the cylinder capacity; and this steam, when discharged by the second cylinder, would have been expanded to the full cylinder capacity; consequently total work corresponding to this expansion would have been performed, and performed in the second cylinder. But by the "annular ring" method we should have no total work credited to this cylinder. This shows the error of the "annular ring" method.

IV. The Thrust of the Screw.

It may be seen that Mr. Isherwood's argument on this point involves the denial of the possibility of *negative slip*, and not only that, but as I understand him, the denial of the possibility of a difference between *apparent slip* and *real slip*. I hope that the "obfuscation" of my ideas, alluded to by Mr. Isherwood, has not led me to misunderstand the position he takes in this con-

nection. I am not disposed to argue the question at length, for *negative slip* has already been very fully discussed in the pages of this journal, and I fear it is one of those subjects upon which no amount of argument leads to conversion. I will say only this much, that since every ship in motion developes eddies more or less pronounced in its wake, it appears reasonable to suppose that a propeller so situated as to benefit by these eddies has an advantage over a propeller so situated as to receive no benefit, and the acknowledgment of this involves the acknowledgement of the difference between *real* and *apparent* slip; beyond this, the question of *negative* slip is a question of amount only. To those who deny the possibility of *negative slip*, its frequent and continued occurrence in the most carefully conducted trials must be most discouraging.

The additional matter bearing upon the subject of thrust which Mr. Isherwood introduces is, I believe, open to no criticism, but its truth in no wise attests the truth of what precedes it, of which it is quite independent.

In conclusion, I would say that I am not insensible to the great services Mr. Isherwood has rendered to the engineering profession. His experiments have been models of scientific conscientiousness; he has not hesitated to set down with fidelity all the observed results, even where they have appeared to conflict with his own theories. It is only to some of his conclusions that I have ventured to take exception, and only because, as coming from one of his high scientific reputation, they might otherwise pass as accepted teachings. Had they been advanced by one less eminent, my article would not have been written.

VI.

REGISTER FOR SPEED TRIALS.

By Assistant Engineer W. D. Weaver, U. S. Navy.

The apparatus, a description of which follows, was devised to carry into effect Engineer-in-Chief Melville's method of making speed trials.

An ordinary chronograph was first suggested, but the delicate nature of such instruments and the great expense for one suitable for use aboard ship, made its adoption impracticable. While making inquiries about forms of chronographs, it was learned that a number of Morse registers had been fitted with two pens for use as chronographs in college laboratories, and as a result the present apparatus was devised.

While in principle a Morse register, it differs in being more carefully constructed; in having a governing attachment, by means of which the speed of the paper may be varied; in having five pens instead of two, and in the method in which motion is given to the pens by the magnet armatures.

The essential principles of the apparatus are as follows:

By means of a clock train a paper tape is passed above the pens, on which, at each electrical contact, a mark is made by one of the latter.

To an arbor of the train is attached one of three inertia governors, by means of which the speed at which the tape is paid out can be regulated. One of these permits a motion of one-half inch per second, another of one inch, and a third of one and one-half inches per second. Running free, the tape has a speed of about two and one-half inches per second.

In order that the speed may be as regular as possible, a Geneva stop is connected to the barrel of the main spring, so that only

that part of its length may be utilized in which the elastic force is approximately constant.

The five pens are actuated by four magnets. To one of the magnets is led the interrupted current of an electric chronometer, to two others the currents established by contact on the port and starboard shafts respectively, and through the fourth a momentary current is passed at the instant of arriving on and leaving the measured mile course. To the latter armature are connected two pens which work on the two margins of the tape, thus permitting a datum line to be drawn from which are measured the fractional parts of time and revolutions.

The instrument will run out 125 feet of tape with very little variation of speed. This, with the inch-per-second governor, corresponds to a time of 24 minutes. It should be remembered that the speed of the tape should not appreciably vary during each period of a second, and that variation with this instrument is infinitesimal.

Owing to the powerful spring and few arbors in the train, the probability of the instrument becoming deranged is small. During a large number of experiments, some of them purposely made very trying, no hitch whatever occurred.

An essential feature is the "piano action" motion of the pens. Through this the time of contact with the tape is infinitesimal, thereby avoiding friction against the tape at a critical point—a defect of the two-pen chronograph.

As only intervals of time, revolutions, and space were required, not the absolute time of passing a range mark or of a given revolution, the pens, for simplicity, are made to mark at a "make" contact; and as the chronometer marks at a "break" contact, a relay must be interposed between it and the register.

Should it be considered that the accuracy within the limits of measurement may be improved by using a "break" contact, it will only be necessary to change the windings of magnets and slightly change the motion to effect this. It is very doubtful, however, if this change would measurably, if at all, increase the accuracy.

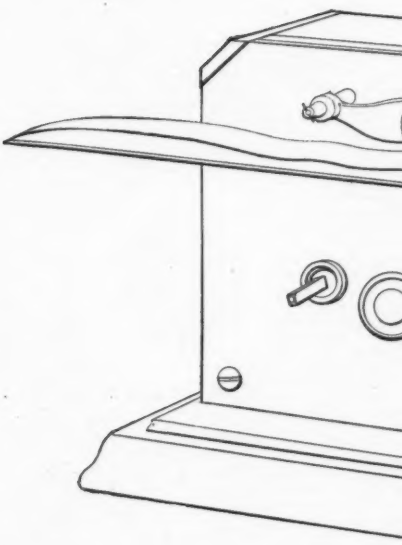
For trial trips it is proposed that there shall be three range observers, one at the bow, another amidships, and a third on the poop. As the ship enters on or leaves the range course, the observer will touch a key and a record will be made on the tape; in addition, there will be a bell button at the bow and stern to notify, from twenty to thirty seconds before the ship enters the course, the one in charge of the instrument that he may set it in motion and switch on the battery.

The diagram of connections shows the manner of setting up the register and will make clear the method of operation.

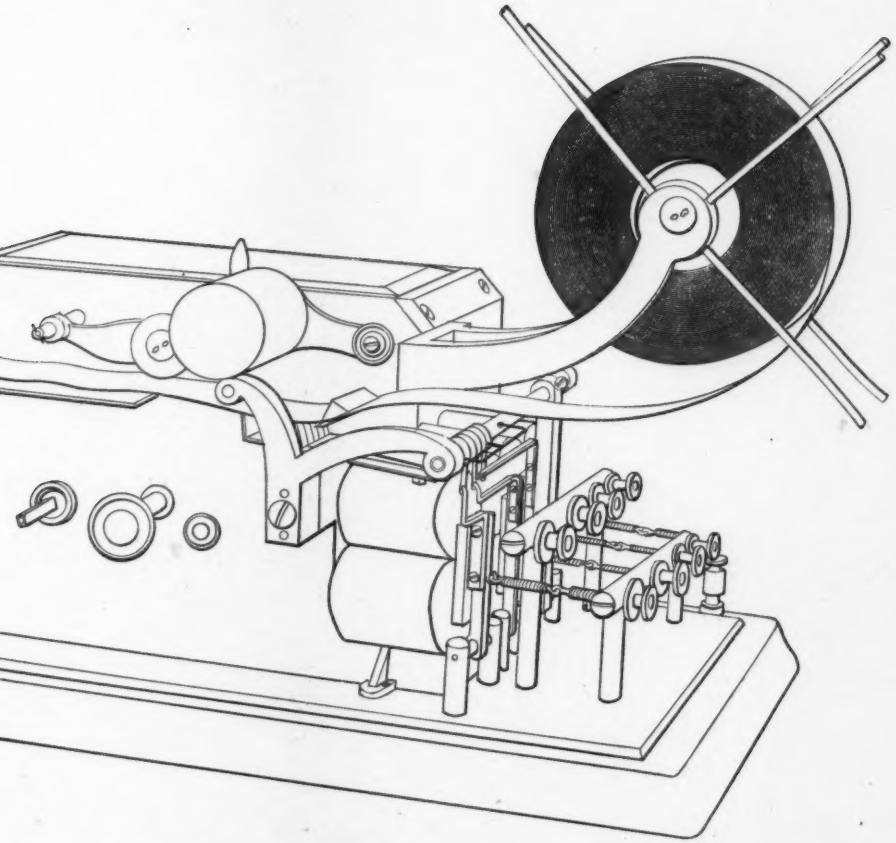
The tape may be measured to give hundredths of a second, and, with a shaft making one hundred and twenty revolutions per minute, fiftieths of a revolution. The marks on the *fac-simile* of record are longer and heavier than on an actual record, and the illustration of the register is inaccurate in several respects, having been made by the engraver from a photograph without proper supervision.

With the instrument made for the Bureau of Steam Engineering were furnished special keyboards, contact brushes, connection board, etc., and drawings and specifications; so that the apparatus could be set up complete for use in a couple of hours after arriving aboard ship, or in the time required for wiring.

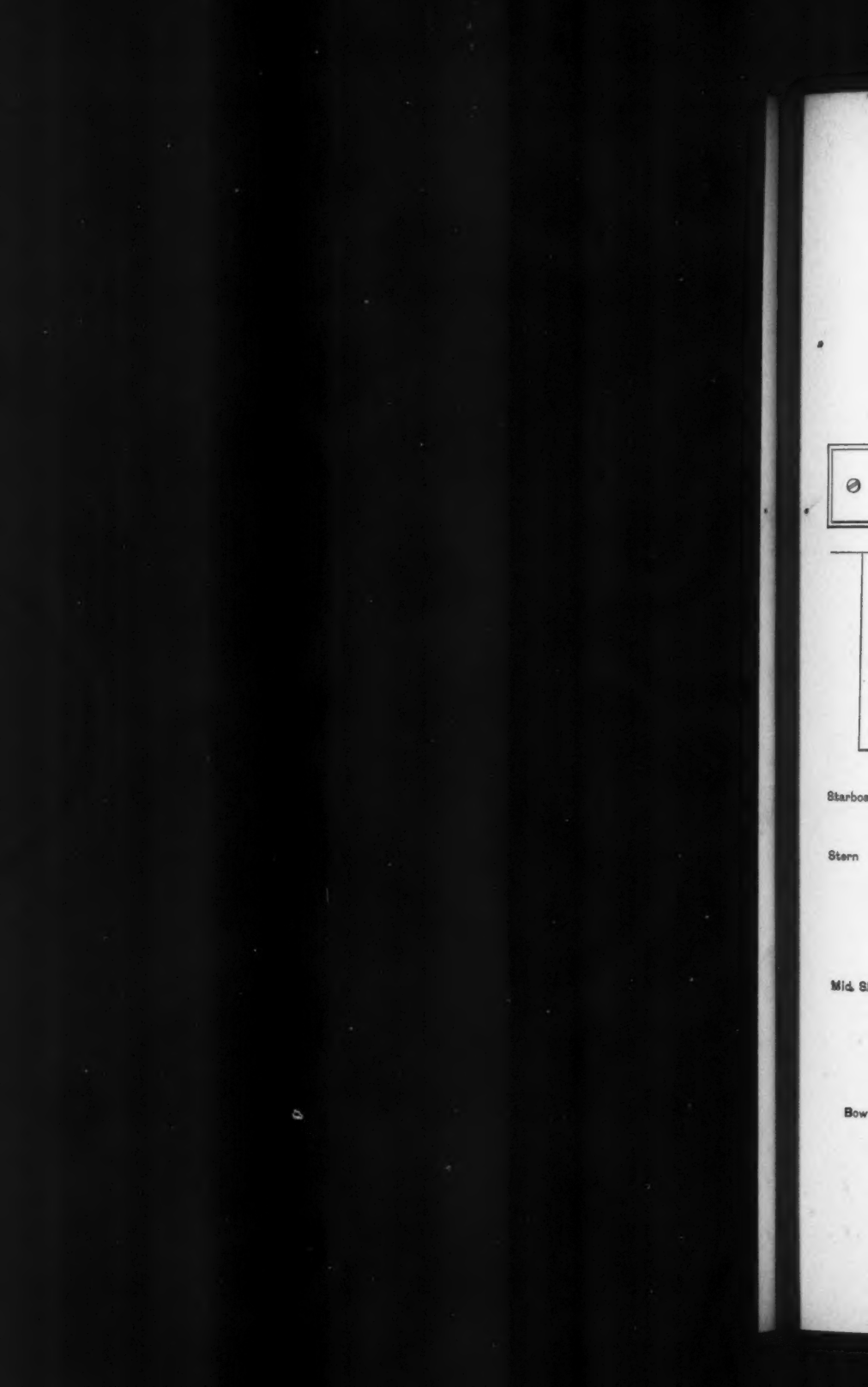
The writer wishes to acknowledge valuable suggestions made by Mr. L. S. Foote, of New York, to whom, besides, the mechanical perfection of the register is largely due.



Range Mark
Port Shaft
Chronometer
Starboard Shaft
Range Mark



FAC-SIMILE OF RECORD.


Starboa
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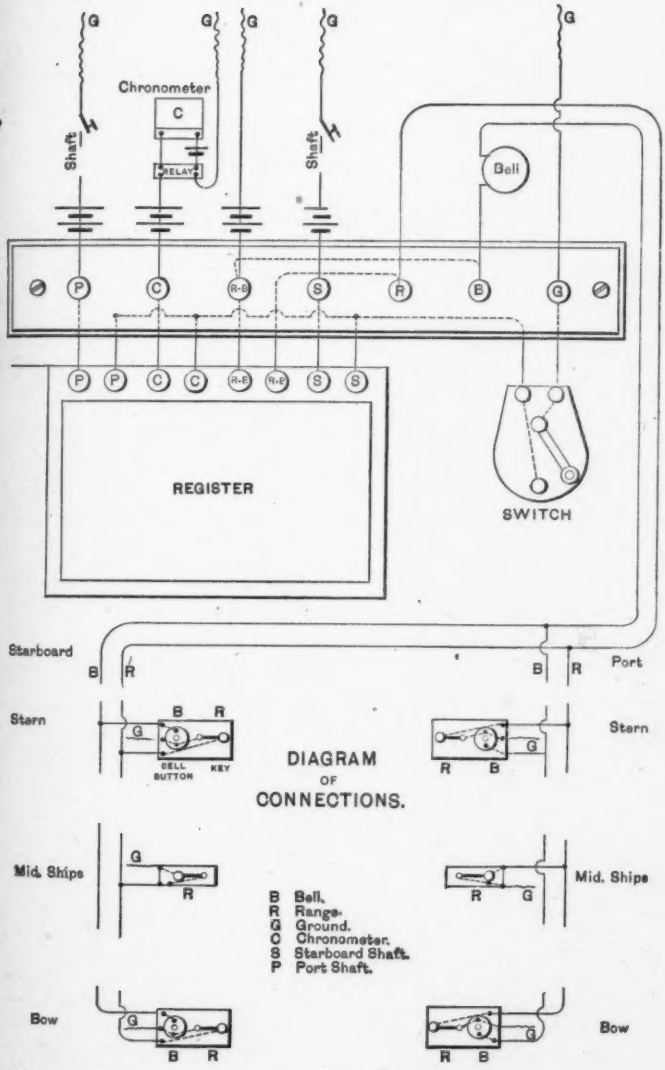
G
Chronometer
C
RELAY
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Bell
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KEY
DIAGRAM
OF
CONNECTIONS.
Mid. Ships
Bow
B Bell.
R Range.
G Ground.
C Chronometer.
S Starboard Shaft.
P Port Shaft.

VII.

CONTRACTORS' FULL POWER FORCED DRAFT TRIAL OF THE U. S. S. *CONCORD.*

By Chief Engineer Robert B. Hine, U. S. Navy.

The four-hour full-power forced-draft trial of the *Concord's* machinery was made by the contractors in Long Island Sound, on Tuesday, January 13th, 1891.

The *Concord* is a twin-screw steel gunboat of 1,703 tons displacement, and was contracted for by N. F. Palmer, Jr., & Co., who built the engines and boilers at the Quintard Iron Works, N. Y., which are owned by them. The hull was sub-contracted to the Delaware River Iron Works at Chester, Pa., and was built there.

Rig: Three-masted schooner. Armament: Six VI-inch B. L. R., and eight rapid-fire machine guns, besides being fitted with tubes for discharging torpedoes.

The following are some of the principal dimensions and data:

Length between perpendiculars	228 feet.
Beam, extreme	36 feet.
Area of immersed midship section	435.3 square feet.
Depth of hold from top of main deck beams to top of floors	18 feet 9 inches.
Draught of water forward 12′ 10″ Draught of water aft 15′ 7″	Mean, 14 feet 2½ inches.
Ratio of length to breadth	6.33 to 1.
Displacement per inch at L. W. L	13.72 tons.
Area of L. W. L	5,765 square feet.
Coefficient of fineness on midship section	.6005
Coefficient of fineness on L. W. L	.7024
Sail area	6,352 square feet.
Transverse meta-center above center of gravity	1.5 to 2 feet.
Moment to alter trim one inch	160 foot tons.
The contract I. H. P. to be developed during the four-hour trial, including blowing, steering, and dynamo engines, and all pumps in use, was	3,400.

Normal capacity of coal bunkers	200 tons.
Maximum capacity of coal bunkers	400 tons.
Displacement on trial	1,740.
Coefficient of fineness on extreme dimensions	.4973

Engines.—There are two horizontal, direct-acting, triple-expansion engines, working at 160 pounds boiler pressure.

The engines are placed one forward of the other, each one being in a separate water-tight compartment. The forward engine drives the port, and the after one the starboard propeller. There is in each engine room an independent air, circulating and bilge pump. The circulating pump is a horizontal Davidson pump, double-acting, which works two vertical single-acting air pumps from its crosshead by means of a rocker arm and overhead beam. A double-acting horizontal bilge pump is worked by an arm from the crosshead.

There is one composition cylindrical condenser in each engine room, with the condensing water passing through the tubes. The valves on the main engines are single-ported piston valves, there being one for each high and intermediate pressure cylinder, and two for each low pressure cylinder. They are worked by Marshall's valve gear, the high and intermediate pressure valves being operated direct, and the low pressure by a rock-shaft. The cut-off is varied by changing the position of the link blocks in rocker arms. The crank shafts are of the built-up type, the webs of cranks being shrunk on and keyed to shafts and pins; the crank shafts and pins are of forged steel. There are axial holes through shafts and crank pins. The crossheads are fitted with slipper guides, and the main bearings and cylinders are tied together by the crosshead guides, and a horizontal and diagonal tie rod for each bearing.

The crank pins are oiled by a centrifugal device, which delivers the oil into the axial hole through the crank pin, and from there it passes through holes to the surface of the pin.

The reversing gear consists of a single horizontal steam cylinder with a floating lever to control its valve, and there is no means provided for reversing by hand. The throttle is a double poppet valve. All the principal bearings are faced with Parsons'

anti-friction metal, except the thrust bearings and crosshead slippers, which are faced with magnolia metal. All the shafting has an axial hole.

There is a Williamson steering engine in the after part of the after engine room, its valve being worked from the steering wheels on deck. The main and auxiliary feed pumps are Davidson vertical pumps, there being one main and one auxiliary pump in each fire room. In each engine room there is a horizontal fire and bilge pump, and in the forward engine room one centrifugal bilge pump, a feed tank, and a horizontal feed pump.

The following are some of the principal dimensions and ratios of the machinery:

Number of cylinders, one engine..... 3.
Diameter of H.P. cylinder..... 22 inches.
Diameter of I.P. cylinder..... 31 inches.
Diameter of L.P. cylinder..... 50 inches.
Stroke, common..... 30 inches.
Cranks are set at 120 degrees from each other, and in the following order for *ahead* motion: L.P., I.P. and H.P.
Diameter of piston rods (forged steel)..... $3\frac{3}{4}$ inches.
Pistons are solid, in dished form, and made of cast steel. Each piston is fitted with an adjustable cast iron shoe.
The cylinders are not steam jacketed. The main valves are piston valves.
Diameter of H.P. valves (one for each cylinder)..... 9 inches.
Area of steam port..... 58.58 square inches.
Diameter of I.P. valves (one for each cylinder)..... 15 inches.
Area of steam port..... 100.58 square inches.
Diameter of L.P. valves (two for each cylinder)..... 17 inches.
Area of steam ports (both valves)..... 225.44 square inches.
Area main steam pipe..... 50.26 square inches.
Volume swept by H.P. piston per stroke..... 6.50 cubic feet.
Volume swept by I.P. piston per stroke..... 13.01 cubic feet.
Volume swept by L.P. piston per stroke..... 33.99 cubic feet.
Diameter of reversing cylinder..... 8 inches.
Stroke of reversing cylinder..... 15 inches.
Clearance, H.P. cylinder, per cent..... 25.46
Clearance, H.P. cylinder, in inches of stroke..... 7.63
Clearance, I.P. cylinder, per cent..... 20.14
Clearance, I.P. cylinder, in inches of stroke..... 6.04
Clearance, L.P. cylinder, per cent..... 19.3
Clearance, L.P. cylinder, in inches of stroke..... 5.8
Ratio of net area of H.P. piston to I.P. piston..... 1 to 2.
Ratio of net area of I.P. piston to L.P. piston..... 1 to 2.613

Ratio of net area H.P. piston to L.P. piston	1 to 5.226
Length of connecting rod between centers	64 inches.
Diameters of connecting rod	$3\frac{3}{4}$ and $4\frac{1}{2}$ inches.
Diameter of crank shaft	9 inches.
Diameter of hole in crank shaft	3 inches.
Diameter of crank pin	9 inches.
Diameter of hole in crank pin	3 inches.
Crank webs, length, over all	$30\frac{1}{2}$ inches.
Crank webs, thickness, (except after one of each set of engines)	$6\frac{1}{2}$ inches.
Crank webs, thickness, (after one, each set engines)	$7\frac{1}{2}$ inches.
Diameter of thrust shaft	9 inches.
Diameter of hole in thrust shaft	4 inches.
Diameter of propeller shafts	$9\frac{3}{4}$ and $11\frac{1}{2}$ inches.
Diameter of hole in propeller shafts	4 and 7 inches.
Length of port shafting, excluding crank shaft	79 feet 9 inches.
Length of starboard shafting, excluding crank shaft	58 feet 7 inches.
Length of crank shafts	14 feet 2 inches.

The outboard sections of propeller shafting are cased with composition $\frac{7}{16}$-inch thick. The flange couplings are $18\frac{1}{2}$ inches diameter and $2\frac{1}{2}$ inches thick. They are protected by water-tight composition covers. The thrust bearing is of the collar type, the rings in block being faced with magnolia metal, and there is a stuffing box at each end of the bearing, so that the rings work in an oil bath.

Outside diameter of rings	$12\frac{1}{2}$ inches.
Inside diameter of rings	9 inches.
Number of rings	9.
Thrust surface	531.91 square inches.
Length of condenser, total	8 feet 11 inches.
Diameter of shell	4 feet 6 inches.
Length of tubes (exposed)	7 feet.
Diameter of tubes (outside)	$\frac{5}{8}$ inches.
Number of tubes (one condenser)	2,151.
Cooling surface (tubes, one condenser)	2,463 square feet.
Cooling surface (total, one condenser)	2,485 square feet.

Independent pumping engine.

Diameter of steam cylinder	14 inches.
Stroke (common to steam cylinder, air and circulating pumps, and attached bilge pump)	12 inches.
Circulating pump, double-acting, diameter	18 inches.
Air pumps (two, single-acting), diameter, each	18 inches.
Ratio of volume swept by L.P. piston per stroke to that swept by both air pump pistons per stroke	9.62 to 1.

Screws.—Type, modified Griffiths, with adjustable blades. With the blades in central position, the pitch is 12′ 6″ and the screws are true. Material, manganese bronze.

Pitch adjustable from 11 to 14 feet.

Diameter	10 feet 6 inches.
Number of blades	3.
Pitch as set, at 10 feet diameter	13 feet 3.36 inches.
Pitch as set, at 4 feet diameter	13 feet 1.68 inches.
Pitch as set, mean of 4′, 6′, 8′, and 10′ diameter	13 feet 2.43 inches.
Helicoidal area	26.52 square feet.
Projected area	21.45 square feet.
Disc area	86.59 square feet.
Ratio of pitch to diameter	1 to 1.257.
Greatest width of blade	2 feet 6 inches.
Diameter of hub	2 feet 6½ inches.

Boilers.—There are four boilers of the low cylindrical type, each having three furnaces, with one combustion chamber, which is divided into three parts by a transverse arch of fire brick, and a longitudinal wall built on the crown of the arch and between the nests of tubes. There is a hanging bridge wall in each of the divisions of combustion chamber. The boilers are placed in two separate water-tight compartments. The Kafer system of forced draft is used. There are two blowers in each fire room, drawing air from the fire rooms, and discharging it into ducts which are led to the ash-pit fronts and door frames. One of the blowers in after fire room is arranged to draw air from the engine room or fire room as may be desired.

There is one smokepipe for the four boilers, the uptakes for each pair uniting above the water-tight bulkheads between the fire rooms.

The following are some of the dimensions of the boilers:

Length	17 feet 9 inches.
Diameter	9 feet 9 inches.
Number of furnaces in each (corrugated)	3.
Material of boilers	Steel.
Diameter of furnaces, maximum	41 inches.
Diameter of furnaces, minimum	37 inches.
Length of grate	5 feet 9 inches.
Outside diameter of tubes	2¼ inches.
Material of tubes	Iron.
Length of tubes between tube sheets	6 feet 11 inches.
Number of common tubes (one boiler)	370.
Thickness of common tubes	No. 11 B. W. G.

Number of stay tubes (one boiler)..... 68.
Thickness of stay tubes..... No. 6 B. W. G.
Total number of tubes (one boiler)..... 438.
Depth of combustion chamber..... 45 inches.
Width of combustion chamber.....8 feet 10 inches.
Height of combustion chamber above upper grates.....20 inches.
Height of combustion chamber above lower grates..... 50 inches.
Thickness of shell plates..... $\frac{1}{16}$ inch.
Thickness of tube sheets..... $\frac{9}{16}$ inch.
Thickness of front head..... $\frac{5}{8}$ inch.
Thickness of back head above tube sheet..... $\frac{3}{4}$ inch.
Thickness of combustion chambers..... $\frac{9}{16}$ inch.
Thickness of furnaces..... $\frac{1}{2}$ inch.
Riveting: Shell, longitudinal seams, double, with double butt straps; circumferential seams, double, with lap joints.
Diameter of rivets..... 1 inch.
Diameter of fore and aft braces..... $2\frac{1}{4}$ inches.
Number of fore and aft braces..... 15.
Load on safety valves..... 168 pounds.
Boilers tested to..... 240 pounds.
Tube-heating surface (one boiler)..... 1,784.52 square feet.
Plate-heating surface (one boiler)..... 267.90 square feet.
Total heating surface (one boiler)..... 2,052.42 square feet.
Grate surface, (one boiler)..... 55 square feet.
Area through tubes, (one boiler)..... 9.42 square feet.
Water surface (one boiler)..... 149.5 square feet.
Steam room (one boiler)..... 327.79 cubic feet.
Capacity of furnaces and combustion chambers above fire grates..... 162 cubic feet.
Weight of one boiler, empty..... 26.68 tons.
Weight of water in one boiler..... 14.98 tons.
Weight of water and boiler..... 41.66 tons.
Diameter of smokepipe..... 7 feet.
Height of smokepipe above lower grate.....57 feet 6 inches.

TOTAL FOR THE FOUR BOILERS.

Grate surface..... 220 square feet.
Heating surface, tube, 7,138.08 square feet. } Total..... 8,209.68 square feet.
Heating surface, plate, 1.071.6 square feet. }
Area through tubes..... 37.68 square feet.
Capacity of furnace and combustion chambers above fire grates..... 648 cubic feet.
Capacity of steam room..... 1,311.16 cubic feet.
Area of water surface..... 598 square feet.
Area through smokepipe..... 37.2 square feet.
Ratios:
Tube H.S. to G.S..... 32.45 to 1.

Plate H.S. to G.S. ..... 4.87 to 1.
Total H.S. to G.S ..... 37.32 to 1.
Area through tubes to G.S. ..... 1 to 5.8
Area of smokepipe to G.S. ..... 1 to 5.9
Water surface to G.S. ..... 2.72 to 1.
Steam Room to G.S. ..... 5.96 to 1.
Capacity of furnace and combustion chambers above fire grates to G.S. ..... 2.95 to 1.
Weight of boilers and water per 100 square feet of H.S. (in tons) ..... 2.21.
Fans, number ..... 4.
Fans, diameter of ..... 4 feet 6 inches.
Type of fan engine, Brotherhood's three-cylinder.
Diameter of cylinders ..... 5 inches.
Stroke of pistons ..... 5 inches.
There is a separator in steam pipe to each set of engines.
Total weight of machinery is (including water in boilers and condensers) ..... 354.77 tons.

Performance.—The *Concord* left the wharf at the foot of Eighth street, East River, at 11.10 A. M., January 12th, 1891, with steam on all boilers, using natural draft, and proceeded to Long Island Sound, where a preliminary run with forced draft was had for about three hours, after which, at 3.55 P. M., the ship anchored off City Island.

On the following morning the anchor was weighed at 9 o'clock, and at 9.30 the official horse-power trial was commenced. During the trial the machinery worked well, although there was considerable leakage of steam from spring stuffing boxes of H.P. cylinders and valve stems. No water was used on the bearings, and the boilers showed no signs of priming. Indicator cards were taken at the commencement of each period of fifteen minutes during the trial. At 1.30 P. M. the engines were doing better than at the commencement of the trial, and the contractors elected to continue it until 2.15 P. M., which was done, and the trial closed at that time. The period taken for the official trial was from 10.15 A. M. to 2.15 P. M.

The coal used was Pardee's Old Hazleton Mine anthracite, of small egg size, and was of good quality. It was hand picked, and was put up in bags containing 110 pounds each, which afforded an opportunity to determine with accuracy the amount consumed.

At the close of the official trial the blowers were stopped and the ship was run out to sea, off Block Island, to ascertain the behavior of the ship in a seaway. There being no such sea on outside as to afford any test of the qualities of the ship, she was headed in again and anchored off New London, Conn., at 6.10 P. M.

On the following day the ship returned to New York, making the run from New London, Conn., to foot of Twenty-sixth street, East River, New York, in about seven hours, using natural draft.

At the commencement of the official trial the draught, forward, was 12 feet 10 inches; aft, 15 feet 7 inches, and the displacement 1,740 tons. At the end of the trial, draught, forward, 12 feet 5 inches; aft, 15 feet 7 inches.

The speed of the vessel was not officially taken. By observation of points passed, the speed over the ground was estimated to average 15.8 knots, with an average tide of more than a knot against her.

SYNOPSIS OF STEAM TRIAL OF THE U. S. S. *CONCORD*, JANUARY 13, 1891.

AVERAGES.

		Forward.	*Aft.*
Steam at boilers (per gauge)		164.3	164.3
		Starboard.	*Port.*
Steam at engines (per gauge)		159.9	159.6
Steam at 1st receiver (absolute)		79.54	77.01
Steam at 2d receiver (absolute)		29.04	28.50
Vacuum in condenser, in inches of mercury		24.11	24.00
Revolutions of main engines		151.47	153.34
Piston speed, in feet per minute		757.35	766.70
Throttle valve		Wide.	Wide.
Cut-off in fraction of stroke from beginning.	H.P.	.*7925	.*7925
	I.P	.78	.78
	L.P.	.78	.78
Temperatures in degrees Fahrenheit.	Engine rooms, average	89.3°	
	Fire rooms, average	91.4°	
	Injection water	36°	
	Discharge water	105°	
	Feed water	139.9°	
Air pressure at ash pits, in inches of water		2.292	
Revolutions of blowers		482.5	512.8
Strokes of air-pump engines		102.8	101.8

*7 from cards.

		Forward.	*Aft.*
H.P. cylinder.	Mean pressure	53.856	53.13
	I.H.P	464.726	464.016
I.P. cylinder..	Mean pressure	30.432	28.759
	I.H.P	524.723	502.036
L.P. cylinder.	Mean pressure	15.362	15.011
	I.H.P.	688.698	670.104
Collective I.H.P. (each main engine)		1,678.147	1,636.156
Collective I.H.P. (both main engines)		3,314.303	
I.H.P. of air and circulating pump engines		27.328	
I.H.P. of blowers		34.774	
I.H.P. of feed pumps		17.624	
I.H.P. of dynamo engine (estimated)		8.	
I.H.P. of steering engine (estimated)		2.50	
Aggregate collective I.H.P. of the main engines and auxiliaries		3,404.529	
Indicated thrust (I.H.P. of main engines only)		27,692	26,670
Indicated thrust per square foot of developed area of propellers (26.52 square feet), pounds		1,044.2	1,005.6
Indicated thrust per square inch of thrust rings, pounds		52.06	50.14
I.H.P. per square foot of G.S.		15.475	
I.H.P. per ton of boilers, water and machinery		9.6	
Square feet of total H.S. per I.H.P		2.41	
Cubic feet swept per I.H.P. per minute by the L.P. pistons		6.136	6.293
Ratio betwen the volume swept per minute by the L.P. piston in double strokes to that of both air-pump pistons in single strokes		28.3 to 1	28.6 to 1
Coal consumed, pounds per hour		9,402.11	
Coal, pounds per square foot of G. S		42.737	
Coal, pounds per hour per I.H.P. (collective, all machinery in operation)		2.762	
Coal, pounds per hour per I.H.P. (collective, main engines, air and circulating pumps)		2.814	
Refuse, pounds per hour		349.47	
Steam, in pounds per I.H.P. for main engines and auxiliaries from cards, S		15.274	
Steam, in pounds per I.H.P. for main engines and auxiliaries from cards, P		14.978	
Water evaporated per square foot of H. S		6.2727	
Water evaporated per pound of coal		5.477	
I.H.P., collective, all machinery in operation during trial, per square foot of G.S		15.475	
I.H.P., main engines, air and circulating pumps, per square foot of G.S		15.189	
	MAXIMUM.		
Steam at boilers (per gauge)		169.	169.

	Starboard.	*Port.*
Steam at engines (per gauge)	165.	165.
Steam at first receiver (absolute)	82.	79.7
Steam at second receiver (absolute)	30.	29.7
Vacuum, in inches of mercury	24.8	25.0
Throttle	8.	8.
Revolutions, main engines	153.1	154.9
Piston speed in feet per minute	765.5	774.5
I.H.P. of H.P. cylinder	482.601	479.645
I.H.P. of I.P cylinder	541.606	538.281
I.H.P. of L.P. cylinder	704.858	675.346
I.H.P. aggregate, each main engine	1,729.065	1,693.273
I.H.P. aggregate, both main engines	3,422.338	
I.H P., air and circulating pump engines	13.407	14.733
I.H.P., blowers	34.774	
I.H.P., feed pumps	17.624	
I.H.P., dynamo (estimated)	8.000	
I.H.P., steering engine (estimated)	2.5	
I.H.P., both main engines and auxiliaries	3,513.376	
I.H.P. per square foot of grate surface	15.97	
I.H.P. per ton of boilers, water, and all machinery	9.903	
Cubic feet swept per I.H.P. per minute by the L.P. piston...	6.19	6.22

The indicator cards shown are for the period when the maximum horse-power was developed, and the mean pressures have been corrected for errors in the indicator springs. I am indebted to Assistant Engineer Albert Moritz for the combined cards shown.

Scale: 80 Lbs.

Scale: 40 Lbs.

Scale: 80 lbs.

Scale: 80 lbs.

U. S. S. CONCORD.

Scale: 80 lbs.

High Pressure Cylinder.

Outboard.

Mean effective press. = 53.538

Inboard.

Mean effective press. = 50.51

Scale: 40 lbs.

Intermediate Pressure Cyl.

Outboard.

Mean effective press. = 28.43

Inboard.

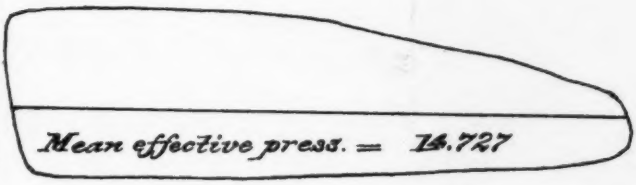
Mean effective press. = 27.93

Scale: 16 lbs.

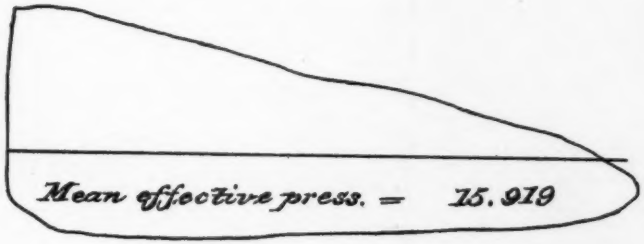
ORD. Port Engine.

Scale: 16 lbs.

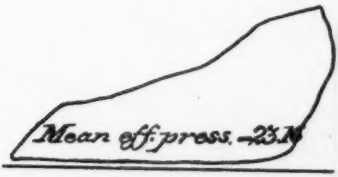
Low Pressure Cylinder. Outboard.



Inboard.



Outboard.



After Starbd Blower.

Scale: 40 lbs.

Revolutions: 490.

U. S. S. CONCORD.

Scale: 80 lbs.

High Pressure Cylinder.

Outboard.

Mean effective press. = 55.227

Inboard.

Mean effective press. = 51.704

Scale: 40 lbs.

Intermediate Pressure Cyl.

Outboard.

Mean effective pressure = 30.637

Inboard.

Mean effective press. = 29.48

Scale: 16 lbs.

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ORD. Starboard Engine.

Scale: 16 lbs.

Low Pressure Cylinder.

Outboard.

Mean effective pressure = 15.276

Inboard.

Mean effective press. = 17.203.

Outboard

Mean eff. press. = 27.32

Mean eff. press. = 29.361

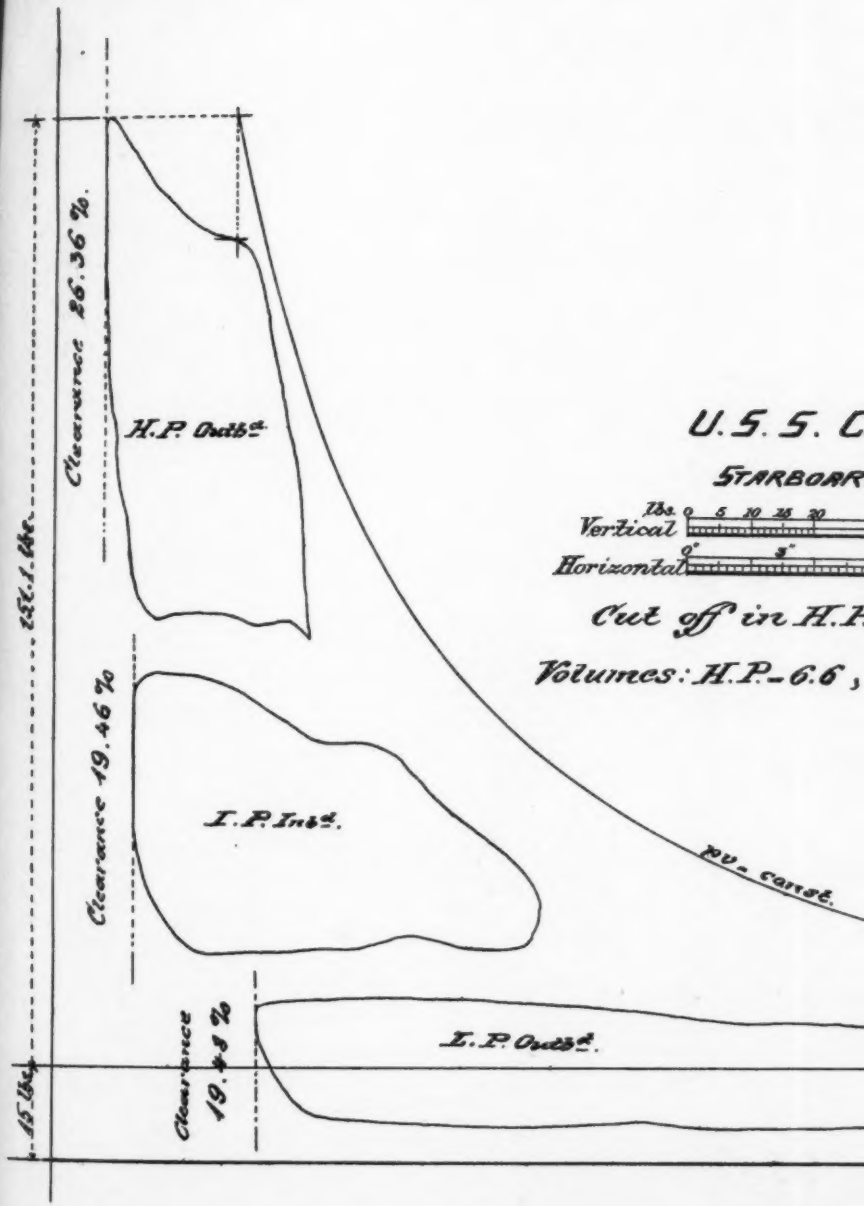
Inboard

Outboard.

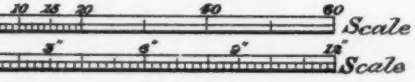
Air and Circulating Pump.

Scale: 16 lbs.

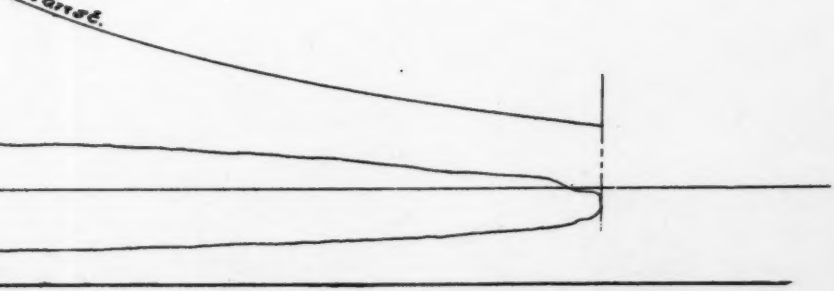
Port Engine Room.

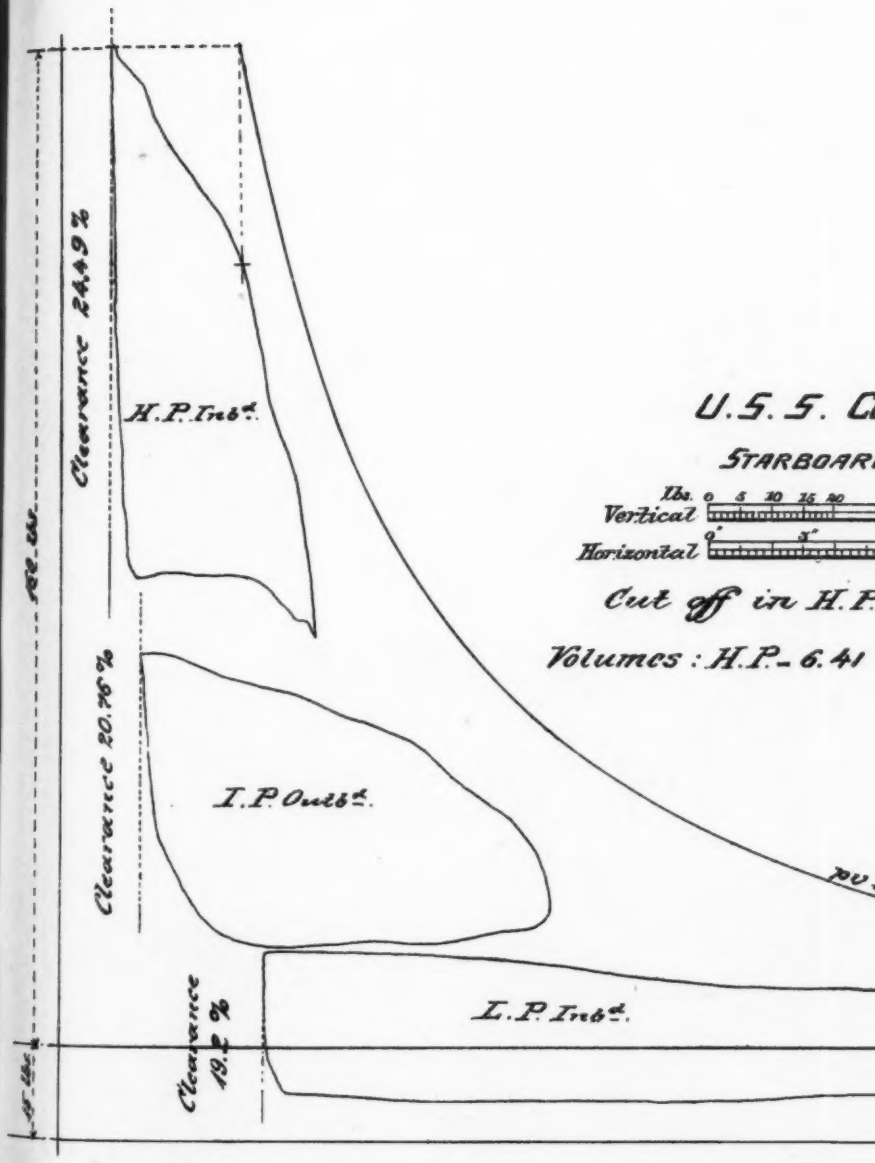

Clearance 26.36 %.
H.P. Outbd.
Clearance 19.46 %
I.P. Inbd.
L.P. Outbd.
Clearance 19.43 %
U.S.S. C
STARBOARD
lbs. 0 5 10 15 20
Vertical
Horizontal
Cut off in H.P.
Volumes: H.P.-6.6,
PD - COMPS.

S. S. CONCORD. _

STARBOARD ENGINE.

in H.P. Cylinder .8 Stroke.

H.P.-6.6, I.P.-12.95, L.P.-34.09 cu. ft.




U.S.S. C
STARBOAR
lbs. 0 5 10 15 20
Vertical
Horizontal
0° 3°
Cut off in H.P.
Volumes: H.P.- 6.41
Clearance 24.9 %
H.P. Ins.
Clearance 20.76 %
I.P. Outs.
Clearance 19.2 %
L.P. Ins.

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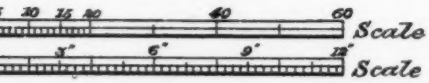
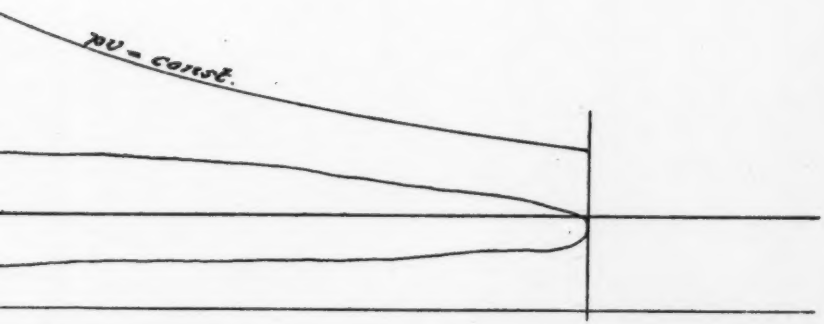
S. S. CONCORD. –

STARBOARD ENGINE.

in H.P. Cylinder . 8 Stroke.

I.P. – 6.41 ; I.P. – 13.10; L.P. – 33.89 sq. ft.



VIII.

THE CONTRACT TRIAL OF THE U. S. S. *NEWARK*.

By Assistant Engineer L. D. Miner, U. S. Navy.

The U. S. S. *Newark* was built by the William Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa. The contract price is $1,248,000.00. The guaranteed horse-power is 8,500; the minimum necesssary for acceptance is 6,000; and the premium is $100 for every horse-power developed above 8,500. The hull was built on the designs of the Department of Construction and Repair. The steel plates were furnished by Carnegie, Phipps & Co., Spanks S. & I. Company, and the London Steel Company, all of Pittsburgh, Pa. The shapes were furnished by the Phœnix Steel and Iron Company, Phœnixville, Pa.

The protective deck extends the whole length of the vessel; the flat top is two inches thick and rises one foot two inches above the mean load line. The inclined sides are three inches thick and extend four feet three inches below the mean load line at frame No. 52. Two bilge keels extend from frame No. 17 to frame No. 60; are 150 feet long, 24 inches deep at frame No. 52 and taper to nothing at the ends. The point of the ram is eight feet below the mean load line and four feet forward of the stem at mean load line. The stem is made of two steel castings scarfed just above the load line. The sternpost is a single steel casting. The double bottom extends from frame No. 31 to frame No. 74, and is divided into fourteen water-tight compartments.

The armament will consist of twelve 6-inch B. L. rifles, four 6-pdr. R. F. guns, four 3-pdr. R. F. guns, two 1-pdr. R. F. guns, three 37-mm. R. C., and four Gatling guns.

There is a poop and topgallant forecastle, two bridges at the same sheer level and one bridge above the conning tower. The

conning tower is at frame No. 30, cylindrcial in form, 3 inches thick, and the peep-holes are 28 feet 6 inches above the load line.

The rig is that of a barque with stump topgallant masts. The lower masts are of steel, the top masts and topgallant masts of wood. The fore and main masts have fighting tops.

There are ten boats. Abaft the forward bridge are a 32-foot sailing launch and a 28-foot cutter on the port side; a 33-foot steam cutter and a 28-foot cutter on the starboard side. Forward of the after bridge a 24-foot cutter is on the port side and a 20-foot cutter on the starboard side. Abaft the after bridge are a 28-foot steam whale boat on the port side and a 29-foot whale boat on the starboard side. On the port quarter is a 29-foot whale boat, and on the starboard quarter a 30-foot gig whale boat.

There are six torpedo tubes; two bow, at frame No. 25; two broadside, at frame No. 52; and two stern, at frame No. 79.

The rudder can be controlled at five different places; in the steering engine room below the protective deck, immediately forward of the tiller room; on the quarter deck; in the conning tower; on the bridge above the conning tower; and in the pilot house immediately abaft the conning tower.

HULL.

Length over all	327 feet 10⅛ inches.
Length on L.W.L	310 feet 10 inches.
Length between perpendiculars	310 feet 10 inches.
Beam, extreme	49 feet 2 inches.
Beam at L.W.L	49 feet 2 inches.
Ratio of length to beam	6.305 to 1.
Depth in hold to gun deck	31 feet 4¼ inches.
Draught, mean	18 feet 9 inches.
Displacement (at load draught)	4098 tons.
Displacement per inch at L.W.L	25 tons.
Area of immersed midship section	800 square feet.
Area of L.W.L. plane	10,500 square feet.
Center of gravity of L.W.L. plane abaft frame No. 52	5 feet 8.4 inches.
Center of buoyancy forward of frame No. 52	1 foot.
Center of buoyancy below L.W.L	7 feet 9 inches.
Center of gravity below L.W.L	.56 foot.

Transverse meta-center above L.W.L.	2 feet 9½ inches.
Longitudinal meta-center above L.W.L.	351 feet 7 inches.
Block coefficient of fineness	.510
Coefficient of fineness on midship section	.588
Coefficient of fineness on L.W.L.	.685
Moment to alter trim one inch	381.69 foot tons.
Wetted surface (by Kirk's analysis)	20,634.
Maximum coal capacity	809.46
Coal below protective deck	314.73
Area of plain sail	11,932 square feet.
Center of effort of plain sail above L.W.L.	61 feet.
Center of effort of plain sail forward frame No. 52	15 feet 10 inches.
Area of rudder	101.25 square feet.
Mean depth below L.W.L.	11 feet 8½ inches.
Maximum angle (about)	43 degrees.

Machinery.—The main engines and boilers are of the contractors' design. There are two main engines, horizontal, triple-expansion, three-cylinder, direct-acting. The forward engine drives the starboard screw, which is right-handed, and the after engine drives the port screw, which is left-handed. The high-pressure cylinder of each engine is forward, the intermediate-cylinder next, and the low-pressure aft. The cranks are placed at angles of 120°, and for ahead motion follow in the order of the cylinders, the high pressure leading. The high and intermediate pressure cylinders have each one single-ported piston valve. The low-pressure cylinders have each two single-ported piston valves. All the valves are worked by Marshall gear, the low-pressure valves being worked from a rock shaft. The steam reversing cylinder is between the intermediate and high-pressure crosshead guides, and is moved by a floating lever. There is no hand reversing gear. A Westinghouse governor may be attached to the reversing engine. The cut-offs are variable between the following limits: for the H.P. cylinder, from .32 to .73; for the I.P. cylinder, from .47 to .73; for the L.P. cylinder, from .5 to .67; they may be altered for each cylinder independently by the usual sliding block in a slot of the reversing arm. There are no independent cut-off gears.

The cylinders are all steam jacketed, the intermediate and low-pressure cylinders through separate reducers. There are two copper exhaust pipes from the high-pressure to the intermediate-pressure cylinder, forming part of the first receiver; and one copper exhaust pipe from the intermediate to the low, forming part of the second receiver. Thus the high-pressure valve takes steam in the center, the intermediate at the ends, and the low-pressure in the center. There are relief valves for connecting the steam nozzles with the exhaust side. The bearing faces of crossheads, the eccentrics, main bearing and crank-pin brasses, and brasses of valve gears, are lined with white metal. Metallic packing is used for piston and valve rods. The main valves have tailrods working in pockets in the chest covers. The pistons are of cast steel, the piston rods of open hearth steel, the connecting rods of wrought iron. For oiling, wipers are used for the crossheads, telescopic gears for the eccentric straps, and wipers and centrifugal oilers for the crank pins, the telescopic gear having proved unsatisfactory for the crank pins when the engines are running above 110 revolutions per minute. The main steam pipes are led as follows: Steam from the two forward boilers passes through a pipe on the port side near the passageway bulkhead. The steam from the two after boilers passes through a pipe on the starboard side well amidships. Stop valves on boilers are automatic closing. Just abaft the forward engine room bulkhead is an automatic stop valve on each main steam pipe, and beyond this is a branch connecting the two pipes, with a stop valve. Steam for the forward engine traverses a portion of this branch and the stop valve on its way to the separator, which is at the bulkhead; thence through a double-beat throttle valve (each opening 9⅝ inches), worked by a system of levers, to the forward H.P. cylinder. The stop valve for the after engine is just beyond the starboard automatic valve. The separator is just abaft the bulkhead dividing the engine rooms. It is apparent from the above that with all boilers and both engines going, steam from any boiler may pass to either engine. Again, any one boiler of the forward or after set may be used on either engine or on both.

Number of cylinders (one engine)	3.	
	Starboard.	*Port.*
Diameter of H.P. cylinder	34 1/16	34 1/16
Diameter of I.P. cylinder	52 3/16	52 3/16
Diameter of L.P. cylinder	76 3/16	76 9/16
Stroke of pistons	3 feet 4 inches.	
Diameter of H.P. valves	18 inches.	
Diameter of I.P. valves	30 inches.	
Diameter of L.P. valves (2)	30 inches.	
Area of H.P. steam ports, maximum opening	Inboard... 81.21 square inches.	
	Outboard.. 106.73 square inches.	
Area of H.P. exhaust ports, maximum opening	Inboard... 148.5 square inches.	
	Outboard.. 141.54 square inches.	
Area of I.P. steam ports, maximum opening	Inboard... 197.7 square inches.	
	Outboard.. 135.92 square inches.	
Area of I.P. exhaust ports, maximum opening	Inboard... 251.24 square inches.	
	Outboard.. 263.5 square inches.	
Area of L.P. steam ports, maximum opening	Inboard... 395.4 square inches.	
	Outboard.. 271.84 square inches.	
Area of L.P. exhaust ports, maximum opening	Inboard... 502.48 square inches.	
	Outboard.. 527.0 square inches.	
Area of H.P. ports, each end	148.5 square inches.	
Area of I.P. and L.P. ports, each end, one valve	263.5 square inches.	
Area cross section of main steam pipe (12½″ diam.)	122.76 square inches.	
Area cross section of H.P. exhaust pipes (2 of 10″ diam.)	157.14 square inches.	
Area cross section of I.P. exhaust pipe (21″ diam.)	346.5 square inches.	
Area cross section of L.P. exhaust pipe (22½″ diam.)	397.77 square inches.	
	Starboard.	*Port.*
Volume swept by H.P. piston per stroke	21.035	21.173
Volume swept by I.P. piston per stroke	49.425	49 221
Volume swept by L.P. piston per stroke	105.53	105.39
Ratio of net area of H.P. piston to I.P.	2.350	2.324
Ratio of net area of I.P piston to L.P.	2.135	2.141
Ratio of net area of H.P. piston to L.P.	5.017	4.978
Clearance in H.P. cylinder, per cent	28.	
Clearance in H.P. cylinder, in inches of stroke	11.2 inches.	
Clearance in I.P. cylinder, per cent	21.5	
Clearance in I.P. cylinder, in inches of stroke	8.6 inches.	
Clearance in L.P. cylinder, per cent	25.6	
Clearance in L.P. cylinder, in inches of stroke	10.24 inches.	
Diameter of piston rods	6 inches.	
Length of piston rods from piston to crosshead	5 feet 3¾ inches (about).	
Diameter of connecting rods	Crank-pin end 6¾ inches.	
	Wrist-pin end 5¾ inches.	
Length of connecting rods between centers	7 feet 2 inches.	
Axis of H.P. cylinders abaft frame 52	Starboard 21 feet 4 inches.	
	Port 47 feet 6½ inches.	

Distance between H.P. and I.P. cylinder axes ..... 5 feet 10 inches.
Distance between I.P. and L.P. cylinder axes ..... 6 feet 6 inches.

The main pistons have each two rings, each $\frac{5}{8}$-inch wide. The cast iron followers have faces $4\frac{1}{4}$ inches deep. Bolted to these and extending nearly half around the cylinder circumference are white metal-lined shoes with bearing faces 7 inches deep. The piston rods are tapered at both ends 1 in 10, have collars at the piston ends and $\frac{1}{4}$-inch shoulders at the crosshead end.

The steel crank shafts are built up and the pins and shafts have axial holes. The crank and thrust shafts are connected by flanges forged on the shafts and six 3-inch steel bolts. The thrust shaft is connected to the propeller shaft by a cone coupling, keyed, and a screw flange on propeller shaft. This flange is secured to the end of the thrust shaft by six 3-inch steel bolts. The propeller shafts have composition sleeves from hubs to inboard of glands of stern bearings. Steel casings cover the shafts from the skin of the ship to the brackets.

Diameter of crank shafts ..... 13 inches.
Length of forward section of shaft ..... 2 feet 10 inches.
Length of second section of shaft ..... 4 feet 7 inches.
Length of third section of shaft ..... 5 feet 3 inches.
Length of after section of shaft ..... 3 feet 4 inches.
Diameter of hole in forward section ..... 6 inches.
Diameter of hole in second section ..... 5 inches.
Diameter of hole in third section ..... 4 inches.
Diameter of hole in after section ..... 4 inches.
Diameter of crank pins ..... 13 inches.
Length of crank pins ..... 15 inches.
Diameter of hole in H.P. pin ..... 6 inches.
Diameter of hole in I.P. pin ..... 5 inches.
Diameter of hole in L.P. pin ..... 4 inches.
Crank webs, thickness ..... H.P. 8 inches. I.P. 9 inches. L.P. 10 inches.

The outline of crank webs is given by a circle of $12\frac{1}{2}$ inches radius from crank-pin center, a circle of 13 inches radius from shaft center, and circles of 12 inches radius tangent to these.

Length of forward bearing ..... 1 foot $8\frac{1}{2}$ inches.
Length of second bearing ..... 2 feet $5\frac{1}{2}$ inches.
Length of third bearing ..... 2 feet $10\frac{1}{2}$ inches.
Length of after bearing ..... 1 foot 8 inches.

Total length of crank shaft	19 feet 9 inches.
Diameter of line shaft	12½ inches.
Length of line shaft	27 feet 5½ inches.
Diameter of hole	5 inches.
Diameter of thrust shaft	12½ inches.
Length { Starboard	18 feet 5 inches.
Length { Port	19 feet 9 inches.
Diameter of hole	5 inches.
Number of collars	12.
Diameter of shaft in thrust bearing	13¾ inches.
Diameter of collars	18 inches.
Thickness of collars	1⅝ inches.
Space between collars	1½ inches.
Bearing surface (each collar 100.571)	1,206.852 square inches.
Diameter of forward propeller shaft	13 inches.
Length	26 feet 1 inch.
Diameter of hole	6 inches.
Length of sleeve	21 feet 2½ inches.
Diameter of after propeller shaft	14 inches.
Length	36 feet.
Diameter of hole	8 inches.
Thickness of composition sleeve	½ inch.
Length of sleeve	4 feet 10 inches.
Length of bracket bearing	3 feet 9 inches.
Total length of shafting { Starboard	126 feet 2½ inches.
Total length of shafting { Port	100 feet 1 inch.

Measured on bulkhead between engine rooms:

Distance center line of ship to center line of both shafts ..... 9 feet 3 inches.

Shafts parallel to center planes.

Measured on forward bulkhead, forward engine room:

Distance base line of ship to center of starboard shaft ..... 10 feet 2 inches.

Measured on after bulkhead of after engine room:

Distance base line of ship to center of same shaft ..... 9 feet 1¼ inches.

The air and bilge pumps, the fire and water service pumps, and the circulating pump of forward engine room are placed on the port side in the order named from forward. These pumps for the after engine are placed on the starboard side in the order named from aft.

The air and bilge pumps are worked by a two-cylinder compound engine taking steam from the auxiliary steam pipe. There are two single-acting, vertical air pumps, worked from the engine cranks. They deliver water to the feed tank in after starboard corner of forward engine room. The bilge pumps are two ver-

tical plunger pumps, worked by cranks on the end of the air-pump engine shaft. They draw water through a Macomb strainer from the main drainage pipes, and deliver it into the main delivery valve chamber.

The fire-and-water service pump is a horizontal Davidson, 8 and 5 by 12 inches stroke.

The circulating pump is centrifugal, and is driven by a horizontal direct-acting engine.

Diameter of air pumps	24 inches.
Stroke	18 inches.
Volume swept per stroke	9.424 cubic feet.
Ratio of volume swept by L.P. piston per stroke to air-pump buckets per stroke	11.19
Diameter high-pressure cylinder	11 inches.
Diameter low-pressure cylinder	18 inches.
Stroke of both	18 inches.
Diameter of bilge pump	4 inches.
Stroke of bilge pump	18 inches.

Diameter of fans of circulating pump	2 feet 8 inches.
Width of fans of circulating pump	5½ inches at hub; 2¾ inches at tips.
Diameter of engine cylinder	10 inches.
Stroke of engine piston	6 inches.
Diameter of suction and discharge	15 inches.

The main condensers are immediately above the circulating and fire-and-water service pumps. The tubes are horizontal, extend fore-and-aft, and can be easily examined and withdrawn. The shells are of cast brass, of three sections each four feet long, with end water chambers. The water circulates through the tubes.

Length over all	14 feet.
Diameter of shell inside	5 feet 8 inches.
Thickness of shell	½ inch.
Length of tubes exposed	12 feet.
Diameter of tubes (thickness 20 B.W.G)	⅝ inch.
Number of tubes, one condenser	3,186.
Cooling surface, one condenser	6,257.304 square feet.
Ratio of cooling surface to heating surface of boilers	1 to 1.3383

The auxiliary condenser is in the forward starboard corner of the forward engine room. The tubes are vertical, with circulat-

ing water outside. The air pump is vertical, single acting, and the circulating pump centrifugal. Both are driven by a single steam cylinder placed vertically over the air pump.

Number of tubes	841.
Length of tubes exposed	4 feet 9 inches.
Outside diameter	$\frac{3}{4}$ inches.
Cooling surface	784 square feet.
Diameter of air pumps	$7\frac{1}{2}$ inches; stroke 7 inches.
Diameter of fans	24 inches; width $2\frac{3}{4}$ inches.
Diameter of steam cylinder	$6\frac{3}{4}$ inches; stroke 7 inches.

The turning engine is near the after bulkhead of the after engine room. Its shaft is horizontal, and through worm gears it works vertical shafts at each end. These vertical shafts have sliding worms which may engage wheels on the main engine shafts.

The propellers were cast by B. H. Cramp & Co., of Philadelphia, Pa. They are of manganese bronze, three-bladed, type, modified Griffiths. The blades are secured to the boss by seven $3\frac{7}{8}$ inches naval brass bolts, and the pitch is adjustable from 17 feet 6 inches to 20 feet 6 inches. The boss is secured to the shaft by a steel key and a nut on the end of the shaft. The taper of the shaft in the boss is $2\frac{19}{32}$ inches, in diameter, in a length of 2 feet $9\frac{1}{2}$ inches. The blades were made to a pitch of 19 feet.

Number of blades	3.
Diameter	14 feet 6 inches.
Pitch as set	18 feet 11.7 inches.
Greatest width of blade (3 feet 11 inches from center)	4 feet.
Length of blades from flange	$63\frac{1}{4}$ inches.
Helicoidal area	52.776 square feet.
Projected area	40 square feet.
Disc area	152.891
Pitch as set ÷ diameter	1.308
Helicoidal area ÷ disc area	.345
Projected area ÷ disc area	.261
Disc area of both propellers ÷ area immersed midship section	.394
Immersion of center at mean draft of 18 feet 9 inches	11 feet 4 inches.
Diameter of hubs	3 feet $11\frac{1}{2}$ inches.

The Boilers.—There are four cylindrical, steel, double-ended, six-furnace boilers placed in two water-tight compartments. The

opposite furnaces in each boiler have a common back connection. The furnaces are corrugated. The circumferential seams are lapped and double zig-zag rivetted. The longitudinal seams are butt-strapped and double zig-zag rivetted. Each boiler is provided with two Weir hydrokineters, two spring safety valves on its forward end, one main stop valve, automatic, on the after head, one auxiliary automatic stop valve on the main stop valve chamber, one stop valve on top about the center of the length for dynamo engines, one main feed check, one auxiliary feed check, and one bottom and one surface blow valve. There are two cylindrical smoke pipes with an annular casing about each.

Length of boiler	19 feet 4½ inches.
Diameter outside	13 feet 6 inches.
Number of furnaces in each	6.
Diameter of furnaces, greatest external	3 feet 11 inches.
Diameter of furnaces, least internal	3 feet 7 inches.
Depth of corrugations	1½ inches.
Length of grate	6 feet.
Number of ordinary tubes	604.
Number of stay tubes	132.
Thickness of tubes	ordinary, 12 B.W.G.; stay, 7 B.W.G.
Diameter of tubes, outside	2½ inches.
Length between tube sheets	7 feet 4½ inches.
Spacing of tubes, horizontally	3¾ inches.
Spacing of tubes, vertically	3½ inches.
Depth of combustion chamber	4 feet 6 inches.
Greatest width, central	2 feet 9 inches.
Greatest width, wing	4 feet 4½ inches.
Height of combustion chamber above back end of upper grate	5 feet 8 inches.
Height of combustion chamber above back end of lower grate	7 feet 10 inches.
Capacity of furnaces and combustion chambers above grates and bridge walls	502 cubic feet.
Thicknesss of shell plates	$1\frac{3}{32}$ inches.
Thickness of heads	$\frac{7}{16}$ inch.
Thickness of tube sheets	$\frac{9}{16}$ inch.
Thickness of furnaces	$\frac{17}{32}$ inch.
Thickness of combustion chamber plates	$\frac{9}{16}$ inch.
Diameter of rivets	1¼ inches.
Number of through steel braces above tubes	20.
Diameter	2¾ inches.
Number of through braces below tubes	14.
Diameter	1¾ inches.
Diameter of braces from heads to back tube sheets, both ends	8.

Diameter ..... 1¾ inches.
Number of braces from top of combustion chamber to shell ..... 56.
Diameter ..... 1¾ inches.
Spacing, lengthwise ..... 6½ inches.
Spacing, across ..... 14 inches.
Spacing of braces above tubes, vertical ..... 13 inches.
Spacing of braces above tubes, horizontal ..... 14 inches.
Spacing of stay bolts, vertical ..... 6⅜ inches.
Spacing of stay bolts, horizontal ..... 6⅜ inches.
Tube heating surface ..... 3,612.84 square feet.
Combustion chamber heating furface ..... 269.75 square feet.
Furnace heating surface ..... 302.379 square feet.
Total for one boiler ..... 4,184.969 square feet.
Grate surface, one furnace ..... 22.5 square feet.
Grate surface, one boiler ..... 135 square feet.
Area through tubes ..... 20.45 square feet.
Area above bridge wall ..... 3.06 square feet.
Air space through grate ..... 50 per cent. of grate, about.
Water surface 6½ inches above upper tubes ..... 227.4 square feet.
Steam space at above water level ..... 513.65 cubic feet.
Diameter of main stop valve ..... 9 inches.
Area through dry pipe holes (about) ..... 127 square inches.
Diameter of each safety valve (double safety valve on each boiler) ..... 5½ inches.
Hood on safety valve ..... 160 pounds.
Test pressure on boilers ..... 240 pounds.
Top of boiler below main load line ..... 9½ inches.
Radius of furnace centers ..... 4 feet 5½ inches.
Centers of furnaces apart ..... 4 feet 4 inches.
Diameter of smoke pipe, each ..... 6 feet 6 inches.
Area of smoke pipe, each ..... 33.18 square feet.
Height of top of pipe above lower grates ..... 62 feet.
Annular space between pipe and casing ..... 4 inches.

There is one auxiliary boiler placed on the protective deck, its axis athwartships at frame No. 53⅛. It is a cylindrical, steel, single-ended, two-furnace boiler with corrugated furnaces. The smoke pipe connects with the after side of the after main smoke pipe.

Length ..... 7 feet 11 inches.
Diameter ..... 8 feet 10⅛⅛ inches.
Least internal diameter of furnaces ..... 2 feet 8 inches.
Length of grate ..... 3 feet 10 inches.
Number of ordinary tubes ..... 72.
Number of stay tubes ..... 24.
Diameter of tubes ..... 3 inches.

Thickness of tubes	ordinary, 12 B.W.G.; stay, 7 B.W.G.
Spacing of tubes	horizontal, 4½ inches; vertical, 4¼ inches.
Length of tubes between tube sheets	5 feet 4⅜ inches.
Tube heating surface	414.69 square feet.
Plate heating surface	140.64 square feet.
Total heating surface	555.33 square feet.
Grate surface, both furnaces	21.4 square feet.
Area through tubes	3.95 square feet.
Area of cross section of smoke-pipe	3 square feet.
Load on safety-valves	160 pounds.
Test pressure	240 pounds.

TOTALS FOR THE FOUR MAIN BOILERS.

Grate surface	540 square feet.
Heating surface, tubes	14,448 square feet.
Heating surface, plates	2,288.5 square feet.
Heating surface, total	16,736.5 square feet.
Area through tubes	81.8 square feet.
Capacity of furnaces and combustion chambers above grates and bridge-walls	2,008 cubic feet.
Steam room	2,053.8 cubic feet.
Area of water surface	1,009.6 square feet.
Ratios:	
Tube H.S. to G.S.	26.76
Plate H.S. to G.S.	4.24
Total H.S. to G.S.	30.99
Area through tubes to G.S	1 to 6.6
Area of smoke pipes to G.S.	1 to 8.14
Water surface to G.S.	1.87 to 1.
Steam room per square foot of G.S.	3.80 cu. feet.
Capacity of furnaces and combustion chambers per square foot of G.S.	3.72 cu. feet.
Weight in boiler compartments per 100 square feet of G.S.	2.37 tons.

The closed fire-room forced-draft system is used and screens are fitted near the ends of the boilers to reduce the space kept under pressure. The pressure is furnished by eight fans, two in each fire room, worked by Brotherhood three-cylinder 5 x 5-inch engines.

There are two main and two auxiliary feed pumps, one of each placed in each central fire room. They are of the Blake vertical, duplex, double-acting type. The main pumps draw only from the feed tanks and deliver into the boilers of their respective compartments. The auxiliaries have nests of valves in both the suction and discharge pipes.

Diameter of pumps	5 inches.
Diameter of steam cylinder	8 inches.
Stroke, common	12 inches.

There are four Williamson two-cylinder $4\frac{1}{2} \times 4\frac{1}{2}$-inch servo-motor ash-hoists, one above each fire room.

The fire rooms are athwartships. There are six athwartship water-tight bulkheads in the boiler and engine space. The forward one is at frame No. 31, the others at frames Nos. 34, $45\frac{3}{8}$, $57\frac{1}{8}$, $65\frac{3}{8}$ and 74. Between the forward two is the largest coal bunker. In the next compartment are placed boilers A and B. In the third compartment are boilers C and D. The after two compartments are occupied by the main engines.

Auxiliary machinery outside of main engine and boiler compartments:—

One "Providence" 15 x 14-inch capstan engine placed on gun deck under edge of topgallant forecastle deck.

One steam winch on after edge of forward boiler hatch.

Two blowers on berth deck for ventilating the ship. They are worked by two-cylinder vertical inverted engines, and so arranged that they can either exhaust air from or press air into the various compartments.

One Williamson steering engine below protective deck, just forward of tiller.

There are fifty-five bunkers in all: thirteen below the protective deck and forty-two on the protective deck, in two rows on each side.

Capacity of lower bunkers	314.73 tons.
Capacity of upper bunkers	456.70 tons.
Total capacity, at 42 cubic feet per ton	771.43 tons.
Reserve bunker	38.03 tons.
Total	809.46 tons.

There is one three-coil distiller, capable of giving 3,000 gallons of potable water per twenty-four hours. It is situated on the berth deck at about No. 45 on the port side outboard.

A Baird evaporator will be placed just abaft the distiller.

The electric light plant consists of three compound Thomson-Houston dynamos of 200 amperes and 80 volts. There are about 483 incandescent lamps of 6,662 total candle power. There will be four projectors of a type not yet decided on, on six mounts, four on carriages, two stationary. The dynamos are driven by 7 x 5-inch two-cylinder, vertical, double-acting Armington-Sims engines, to which they are directly coupled by Brotherhood flexible couplings. Minimum steam pressure required, 80 pounds.

WEIGHTS.

Propelling machinery	632.54 tons.
Water in above machinery	101.44 tons.
Tools, instruments and duplicate parts	19.385 tons.
Miscellaneous machinery	9.65 tons.
Total weight	763 tons.
One main boiler without fittings, empty	45.85 tons.
Water in one boiler	22.32 tons.
Weight of one boiler with fittings and water	80.12 tons.
All main boilers with fittings and water	320.48 tons.
All main boilers, fittings, water, uptakes, smoke pipes, and all machinery in fire rooms	396.4 tons.
Auxiliary boiler without fittings, empty	9.15 tons.
Water in auxiliary boiler	6.16 tons.
Auxiliary boiler with water and fittings	16.19 tons.
Auxiliary boiler with fittings, water, uptake, smoke pipe, and all machinery in fire room	18.28 tons.
Total of main and auxiliary boilers, fittings, water, uptakes, smoke pipes, and all machinery in fire rooms	414.62 tons.
Reciprocating parts:	
H.P	4,514.5 pounds.
I.P	5,528.87 pounds.
L.P	7,686.37 pounds.

TRIAL TRIP.

Two unsuccessful trials of the machinery were made on the 10th and 12th of December, 1890. In both cases the starboard engine was disabled by the breakage of the two bolts which secure the cap of the radius link of the L.P. valve gear.

The bolts were replaced by larger ones, and the valve connecting rod was shortened seven-eighths of an inch, and on the 22d of December, 1890, between 7.25 and 12.25 p. m., a successful four-hour trial was run. This trial showed an indicated horse-

power of $8{,}868\frac{677}{1000}$, as maintained by the main and auxiliary machinery, giving the firm a premium of $36,857.70.

The trial passed off with only one accident, the stoppage of one blower for fifteen minutes. The main boilers were operated under forced draft, mean air pressure, 2.23 inches of water. They showed no signs of foaming, and furnished ample steam. They were quite capable of furnishing more than the engines were able to use. A few small unimportant leaks occurred in and about the furnaces.

The engines worked smoothly and the journals showed no excessive heating. Water was circulated through the brasses and sprayed on the crank pins, but this latter was done as a precaution.

The lubricating apparatus was found to answer every requirement, and means for cooling any heated part ample.

The engines were started and reversed quickly and without difficulty.

The independent air pumps give much trouble in starting and run irregularly, although the cranks have been changed several times and are now at 180°.

Steam was admitted to the engines only through the main stop valve to the high-pressure cylinders.

On December 23d a supplementary trial was run for arriving at the amount of coal burned. This lasted four hours, and but two boilers were used under forced draft. There was no difficulty experienced in getting the coal from the bunkers during the trials.

The fuel used was Pocahontas coal, obtained from the Pocahontas Flat Top Mine, Pocahontas, Tazewell County, Virginia. It was selected lump and clean.

Since the trials the machinery has been carefully examined and found in good condition with the exception of slight and unimportant leaks in boiler fronts and back connections. Several of the cast-iron bridge walls were burned and warped.

The number and rates of pay of persons employed on the trials and the material used are as follows:

4 engineers, at $3.25 per diem	29½ days.
33 oilers, at $3 25 per diem	247 days.
5 water-tenders, at $3.25 per diem	46½ days.
32 firemen, at $3 25 per diem	226½ days.
22 coal passers, at $2.50 per diem	133 days.
6 mess men, at $2.50 per diem	41½ days.
1 storekeeper, at $2.50 per diem	9 days.
103	$2,246 06
397 tons of coal, at $7.32	$2,774 28
468 gallons of lard oil, at 60 cents.	280 80
204 gallons cylinder oil, at 70 cents.	142 80
102½ gallons mineral sperm oil, at 20 cents	20 50
368 lbs. castor oil, at 14 cents	51 52
50 lbs. refined tallow, at 25 cents.	12 50
6 lbs. French chalk, at 20 cents	1 20
5 lbs. sulphur, at 5 cents	25
1 lb. curled hair, at $1.50	1 50
50 feet gum hose, at 30 cents	15 00
15 lbs. graphite, at 25 cents	3 75
321 lbs. of waste, at 10 cents.	32 10
50 lbs. of oat meal, at 5 cents	2 50
36 tin oilers, at 12½ cents.	4 50
36 quart oil feeders, at 10 $\frac{5}{12}$ cents.	3 75
12 coal shovels, at 75 cents	9 00
12 tin hand lamps, at 95$\frac{5}{8}$ cents.	11 50
24 bunker lamps, at 50 cents.	12 00
12 brooms, at 37½ cents.	4 50
12 tin cups, at 5 cents	60
	$3,384 55
	2,246 06
Grand total	$5,630 61

During the horse-power trial, which occurred about fifteen miles off shore, the sea was moderate, and the wind was abeam, except during the turn near the middle of the run.

The turning trials took place in the afternoon of the 22d of December, and resulted as follows:

	Time to put helm over.	Helm angle.	Time to complete circle.	Revolutions.
	sec.		*m. sec.*	
Helm hard astarboard, both screws ahead	15	44°	4 45	90
Helm hard aport, both screws ahead	11	43°	4 07	100
Helm hard aport, port screw ahead, starb. stopped	10	42½°	5 30	84
Helm hard astarboard, starb. screw ahead, port astern	15	43°	4 40	S—90 P—80
Helm hard aport, port screw ahead, half speed, starb. screw astern	14	43°	5 30	S—96 P—72

TRIAL DATA.

Draught at beginning.....Forward, 16 feet; aft, 20 feet 8 inches; mean, 18 feet 4 inches.

Draught at end.....Forward, 15 feet 10½ inches; aft, 20 feet 6½ inches; mean, 18 feet 2½ inches.

Draught, mean, for trial.....Forward, 15 feet 11¼ inches; aft, 20 feet 7¼ inches; mean, 18 feet 3¼ inches.

Displacement ..... 3,980 tons.

Area immersed midship section ..... 775 square feet.

Wetted surface, Kirk's analysis ..... 19,110 square feet.

I.H.P. per 100 square feet wetted surface 46.41

I.H.P. per 100 square feet wetted surface at 10 knots in ratio of 3.5 power 4.905

Average speed (by two Walker Cherub logs) ..... 19 knots.

Slip, mean ..... Starboard, 20.310; port, 19.879

Speed ^{3} × area immersed midship section ÷ I.H.P. 613.06

Speed ^{3} × displacement $^{\frac{2}{3}}$ ÷ I.H.P. 198.62

Pitch of propellers ..... 18 feet 11.7 inches.

STEAM LOG.

	Starboard.	*Port.*
Revolutions of main engines per minute	127.34	126.656
Revolutions, mean	126.998	
Piston speed, feet per minute	849.	844.4
Steam at boilers, gauge	161.87	
Steam at engines, gauge	156.12	152.63
Steam at 1st receiver, absolute	57.43	59.71
Steam at 2d receiver, absolute	16.83	16.64
Vacuum in condenser, inches of mercury	25.97	25.66
Opening of throttle	8.	8.
Cut-off in fraction of stroke, H.P	.756	.720
Cut-off in fraction of stroke, I.P	.760	.720
Cut-off in fraction of stroke, L.P	.620	.650
Double strokes of air-pumps per minute	105.1	125.2
Revolutions of circulating pump	207.9	209.1
Revolutions of blowers per minute	643.5	686.77

	Starboard.	*Port.*
Double strokes of feed pumps	30.	
Temperature, engine rooms	86.1	70.3
Temperature, injection	44.2	45.
Temperature, discharge	103.3	75.3
Temperature, feed	112.4	78.3
Temperature, fire rooms	max., 75.9	mean, 66.3
Air pressure	2.23 inches.	
Mean pressure H.P. inboard	59.601	53.622
Mean pressure H.P. outboard	58.114	52.561
Mean pressure I.P. inboard	25.835	25.015
Mean pressure I.P. outboard	24.023	23.102
Mean pressure L.P. inboard	15.395	14.360
Mean pressure L.P. outboard	14.706	12.753
Indicated horse-power, H.P.	1,353.489	1,223.022
Indicated horse-power, I.P.	1,355.202	1,316.633
Indicated horse-power, L.P.	1,759.664	1,574.291
Aggregate equivalent mean pressure on L.P. cylinders	38.2305	35.2835
Collective I.H.P. of each main engine	4,468.357	4,113.946
Collective I.H.P. of both main engines	8,582.303	
I.H.P. of air pumps	29.97	31.597
I.H.P. of circulating pumps (2)	30.208	
I.H.P. of air and circulating pumps	45.074	46.701
I.H.P. of blowers (8)	117.320	
I.H.P. of feed pumps (4)	52.988	
I.H.P. of auxiliary air and circulating pump	4.130	
I.H.P. of water service pumps (2)	9.228	
I.H.P. of dynamo engine, estimated	8.330	
I.H.P. of steering engine, estimated	2.500	
I.H.P. of above auxiliaries	194.496	
I.H.P. of main engine, air and circulating pumps	4,513.431	4,160.647
I.H.P. of both main engines, air and circulating pumps	8,674.078	
Total I.H.P. of both main engines and all auxiliaries	8,868.572	
Indicated thrust	61,026 lbs.	56,489 lbs.
Indicated thrust per square foot developed area of propeller	1,156.3 lbs.	1,070.4 lbs.
Indicated thrust per square inch of surface of thrust bearing	47.98 lbs.	44.41 lbs.
Cubic feet swept per minute by L.P. piston per I.H.P., main engines only	6.015	6.488
Square feet of cooling surface per I.H.P.	1.3868	1.504
Square feet of total heating surface per I.H.P.	1.887	
I.H.P. per square foot of G.S.	16.42	
I.H.P. per ton of all machinery for propelling	12.08	
I.H.P. per ton of total weight of machinery	11.62	

Velocity of steam per second at maximum opening of valves:	Starboard.		Port.	
	Inboard.	*Outboard.*	*Inboard.*	*Outboard.*
H.P. steam port	153.2	120.3	153.5	120.5
I.P steam port	150.2	221.5	149.3	220.1
L.P. steam port	162.1	237.3	161.0	235.6
H.P. exhaust port	83.8	90.76	83.98	90.9
I.P. exhaust port	118.2	114.2	117.5	113.5
L.P. exhaust port	127.6	122.4	126.6	121.5
Main steam pipe		103.1		
Main exhaust pipe		161.1		

MAXIMUM PERFORMANCE—11.55 A. M.

	Starboard.	*Port.*
Steam at boilers (per gauge)	162	
Steam at engines (per gauge)	157	150
Steam in first receiver (absolute)	75	74.7
Steam in second receiver (absolute)	32.5	31.2
Vacuum in inches of mercury	25.5	25.7
Throttle	Wide.	
Revolutions	129.5	128.7
Piston speed in feet per minute	863.33	858.
Mean effective pressure H.P., inboard	57.516	54.065
Mean effective pressure H.P., outboard	58.810	52.967
Mean effective pressure I.P., inboard	26.673	25.783
Mean effective pressure I.P., outboard	24.958	22.652
Mean effective pressure L.P., inboard	15.829	14.478
Mean effective pressure L.P., outboard	15.106	13.268
I.H.P. of H.P. cylinder	1,361.525	1,252.731
I.H.P. of I.P. cylinder	1,427.022	1,329.334
I.H.P. of L.P. cylinder	1,838.919	1,637.037
Collective I.H.P., each main engine	4,627.466	4,219.102
Collective I.H.P., both main engines	8,846.568	
I.H.P. air and circulating pumps	39.929	42.416
I.H.P. main engine, air and circulating pumps	8,928.913	
I.H.P. blowers	125.424	
I.H.P. feed pumps	53.000	
I.H.P. all other auxiliaries	24.188	
I.H.P. all machinery	9,131.525	
Cubic feet per minute swept by L.P. piston per I.H.P. (main engines)	5.88	6.41
Revolutions of air pumps	105	116
Revolutions of circulating pumps	215	220
Double strokes of feed pumps	32.25	
Revolutions of blowers	642	715.5
Air pressure in fire rooms in inches of water	2.25	
Cut-off H.P.	.76	.72
Cut-off I.P.	.76	.72
Cut-off L.P.	.62	.65

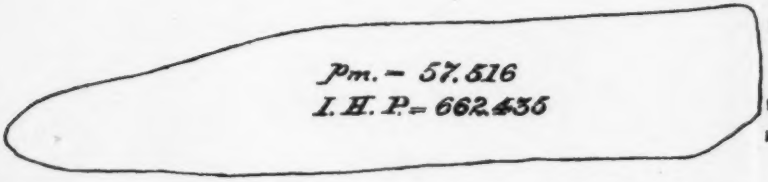
I.H.P. per square foot of G.S.	16.91
Total H.S. + I.H.P.	1.83
I.H.P. per ton of propelling machinery and water	12.44
I.H.P. per ton of all machinery	11.97

COAL CONSUMPTION TRIAL.

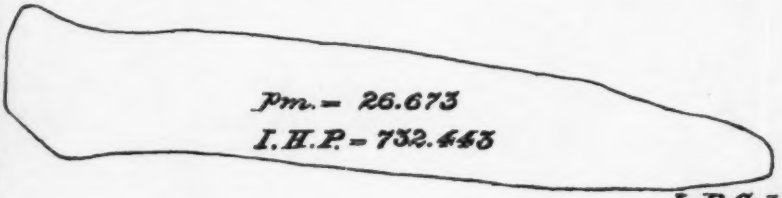
	Starboard.	*Port.*
Steam at boilers (per gauge)	125.5	
Steam at engines (per gauge)	116.	120.
Steam in first receiver	46.6	40.6
Steam in second receiver	20.12	17.7
Vacuum in condenser, in inches of mercury	26.95	26.55
Throttle, holes	2.00	2.47
Cut-off H.P.	.76	.72
Cut-off I.P.	.76	.72
Cut-off L.P.	.62	.65
Revolutions	105.05	102.52
Piston speed, in feet per minute	700.	679.
Mean effective pressure, H.P., inboard	42 502	39.728
Mean effective pressure, H.P., outboard	40.392	42.558
Mean effective pressure, I.P., inboard	15.455	13.348
Mean effective pressure, I.P., outboard	14.861	13.450
Mean effective pressure, L.P., inboard	10.771	7.674
Mean effective pressure, L.P., outboard	10.051	7.066
Horse-power, H.P.	786.74	767.75
Horse-power, I.P.	680.15	585.98
Horse-power, L.P.	1,004.42	692.31
Total	2,471.312	2,046.04
Air and circulating pumps, I.H.P.	33.033	22.152
Collective I.H.P.	2,504.346	2,068.189
Collective of both engines	4,572.535	
Auxiliary air and circulating pump, I.H.P.		3.002
2 fire and water service pumps		3.942
1 main feed pump		11.497
1 auxiliary feed pump		11.497
4 blower engines		67.972
1 dynamo engine (estimated)		8.33
1 steering engine (estimated)		2.50
Total auxiliaries		108.740
Total I.H.P. main engines and auxiliaries		4,681.275
Air pressure		2.25 inches.
Square feet grate surface in use		270.00
Square feet heating surface in use		8,369.94
Vessel's bottom		clean.
Kind and quality of fuel used		bituminous, excellent quality.
Net weight in pounds of fuel burned per hour		10,795.25
Net weight in pounds of combustible burned per hour		10,443.125

U. S. S. NE

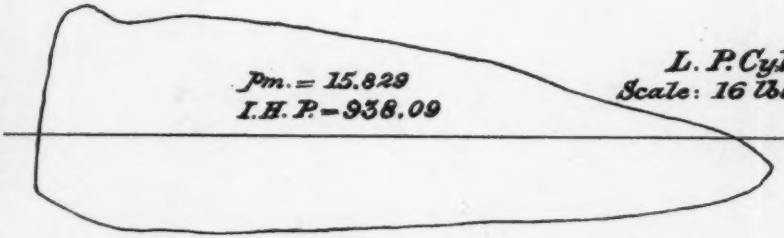
Port Engi



H.P. Cylinder
Scale: 80 lbs. = 1 in



I. P. Cylinder
Scale: 40 lbs. = 1 in



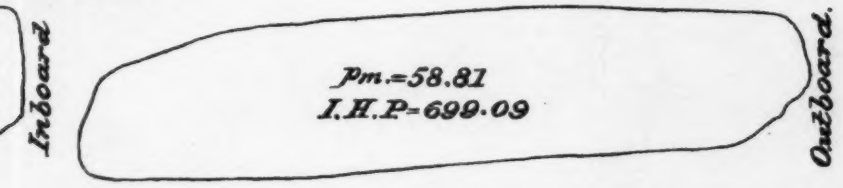
L. P. Cylinder
Scale: 16 lbs. = 1 in

Steam at boilers	162 (per gauge)	Cut
" " engine	157 " "	Thro
" in 1st receiver	75 (absolute)	Revo
" " 2nd "	32·5 "	Vacu

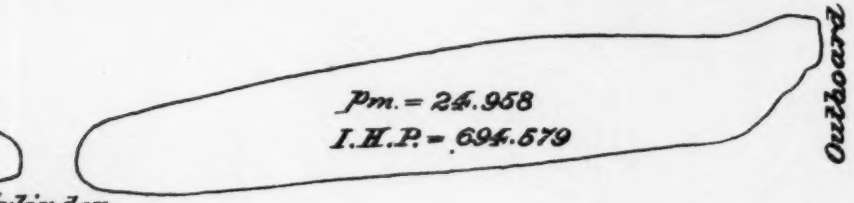
Total I.H.P. of engine 462

. S. NEWARK.

t Engine.

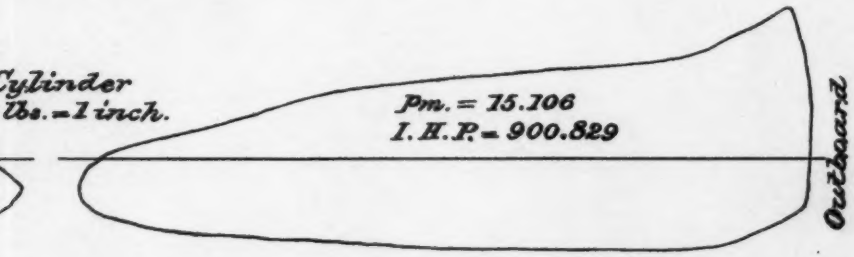


Cylinder
lbs. = 1 inch.



ylinder
lbs.=1inch.

Cylinder
lbs.=1 inch.



) Cut off: H.P. 0.756, I.P. 076, L.P. 0.62

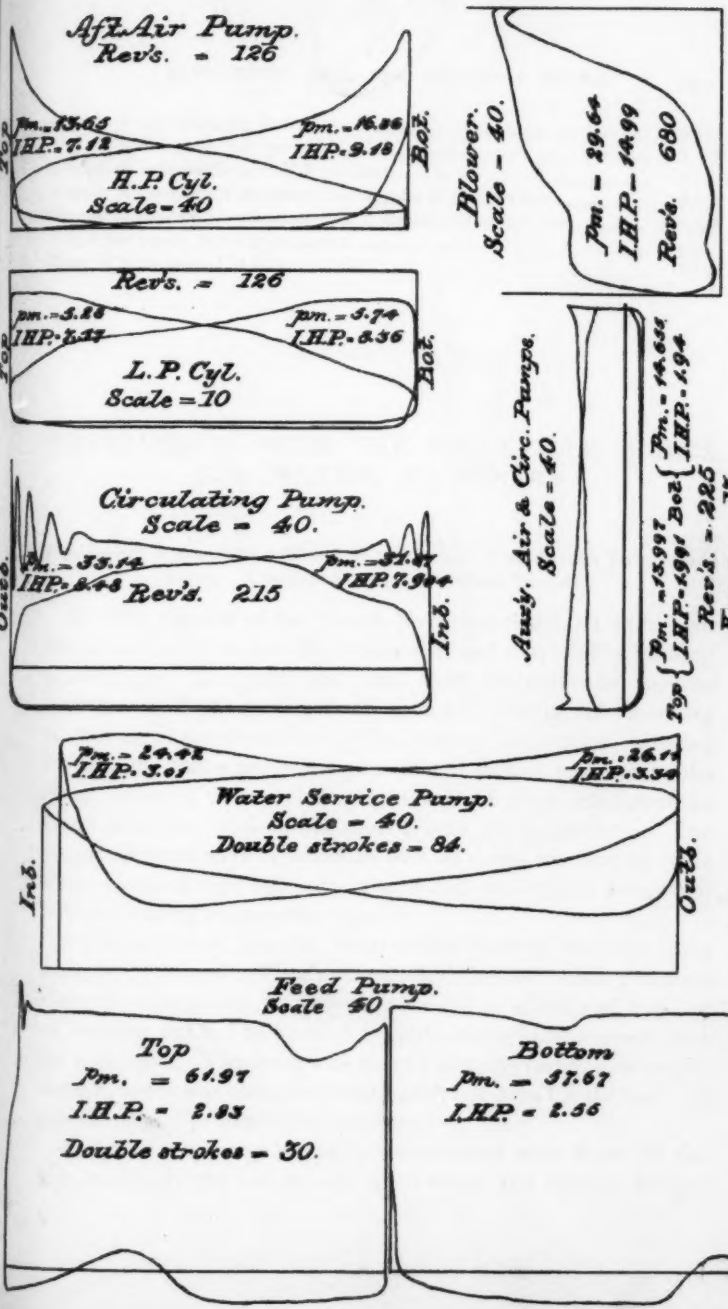
Throttle wide

Revolutions 129.5

Vacuum 25.5

ne 4627.466

For
Ins.
Outs.


Aft Air Pump.
Rev's. = 126
Pm. = 13.65
I.H.P. = 7.12
Pm. = 16.86
I.H.P. = 9.18
Top
Bot.
H.P. Cyl.
Scale = 40
Rev's. = 126
Pm. = 3.26
I.H.P. = 2.17
Pm. = 3.74
I.H.P. = 2.36
Top
Bot.
L.P. Cyl.
Scale = 10
Circulating Pump.
Scale = 40.
Pm. = 33.14
I.H.P. = 8.48
Rev'd. 215
Pm. = 31.47
I.H.P. 7.904
Outs.
Ins.
Blower.
Scale = 40.
Pm. = 29.64
I.H.P. = 14.99
Rev's. 680
Auxly. Air & Circ. Pumps.
Scale = 40.
Top { Pm. = 15.997
I.H.P. = 1.991
Bot { Pm. = 14.654
I.H.P. = 1.94
Rev's. = 225
Vacuum = 13
Pm. = 24.42
I.H.P. = 3.01
Pm. = 26.14
I.H.P. = 3.34
Water Service Pump.
Scale = 40.
Double strokes = 84.
Ins.
Outs.
Feed Pump.
Scale 40
Top
Pm. = 61.97
I.H.P. = 2.83
Double strokes = 30.
Bottom
Pm. = 37.67
I.H.P. = 2.56

Pounds of dry refuse, per hour	356.125
Pounds of fuel per I.H.P. per hour	2.306
Pounds of combustible per I.H.P. per hour	2.23
Pounds of combustible per square foot of grate surface per hour	38.678
Pounds of combustible per square foot of heating surface per hour	1.248
I.H.P. per square foot of grate surface	17.34
Total H.S. in use ÷ I.H.P.	1.788

EXPERIMENTS WITH THE BELLEVILLE BOILER FOR MARINE MACHINERY.

[An abstract from a paper published in the Journal of the Swedish Technological Society. Contributed by C. A. Blomberg, Associate.]

In 1887 a vessel of the French Navy was fitted out with eight Belleville boilers to test their durability and adaptability for long sea voyages. A voyage was made from Marseilles to London and return. Salt water was used to make up losses, which, owing to the use of the steam blast in the chimney, were great. During the run the tubes and other parts of the boilers exposed to the direct action of the heat were examined. A great number of the tubes developed leaks from splits 1¼ to 3¼ inches long. The damaged tubes were repaired as well as could be done by plate strips clamped over the leaky places, and the voyage completed without having to stop the engines.

Upon arrival at London, examination showed that the tubes were covered with scale about one-eighth inch thick, yet, contrary to expectation, there were no signs of change of form or of burning out. The number of splits averaged from one to two for each tube. The cause was sought in faulty material or manufacture, but it was conceded, that had it not been for the scale, no trouble would probably have arisen.

After the return to Marseilles evaporators were fitted on the ship to supply the boilers with fresh water, and certain changes

were made in the internal arrangements of the steam drum or separator. This separator, as originally fitted, may be described as a cylindrical shell with a baffling plate in the interior, concentric with the shell and carried about three-fourths of the way around. The mixture of steam and water rising from the tube elements (a mixture composed of from seventeen to nineteen pounds of water to about two pounds of steam, according to experiment,) enters the annular space so formed and is carried around, whereupon the entrained water is projected to the bottom of the inner shell to flow back into the elements or the sediment collector, while the steam is taken off at the top by the dry pipe. The feed water enters a chamber at the end opposite that at which the water separated out returns to the bottom of the boiler. Through a hole in an otherwise solid diaphragm this chamber communicates with the central portion of the drum. The feed water, entering in fine sprays, is brought in contact with the steam through this hole and is heated. It is known, that when water containing the scale-forming ingredients is heated rapidly to temperatures corresponding with steam pressures of from 120 to 150 pounds, absolute,—about 345°—the scale deposited will be loose and friable, only slightly adhering to the heating surfaces, whereas when the heating takes place slowly, the scale will be hard and adhering.

Now the chamber for heating the feed water as described was faulty in several particulars: It was too small to heat the water rapidly, and it afforded no escape for any air that entered with the water or the steam, the presence of which obstructed the mixing of steam and water and retarded the heating.

The change made in the drum consisted in conducting the feed water directly into the annular space above where the tube elements discharge. It thus passed through the annular space in company with the steam, becoming rapidly heated to the maximum temperature.

The foregoing changes having been made, a second voyage was undertaken, and the boilers were in continuous use for fifty hours. During this run, six of the boilers were fed with fresh water from the evaporators, while the remaining two were fed

with salt water. Upon the return of the vessel, an inspection of the boilers showed them to be in a remarkably good condition. As a matter of course, scale was found in the two boilers that had been fed with salt water, but it was in the form of a powder. A sample of the scale, when dried, formed a hard substance, but could, without trouble, be pulverized between the fingers.

Another thing demonstrated by this latter trial was that the sediment collector functioned satisfactorily. The upward velocity of the water in the sediment chamber does not exceed 12 to 16 inches per second, giving ample time for the deposition of the suspended particles. During the fifty-hour trial a total quantity of 90,000 litres (3,177 cubic feet) of water was pumped into the two boilers using salt water, containing 297.63 pounds of solid matter, yet the examination after the trial revealed the presence of only an insignificant quantity. The sediment was blown from the collector every half hour.

NOTES.

A NEW EXPLANATION OF THE ACCIDENT TO THE *CITY OF PARIS*.

Although accounts and explanations of the accident that befel the *City of Paris* are without number, it may yet be of interest to our readers to read the conclusions arrived at by a German engineer, Mr. O. H. Mueller, who undertook a journey to England for the single purpose of arriving at an explanation, satisfactory to himself, of the causes of the catastrophe and the succession of events during the breakdown. What follows is an abstract of his report in the "*Zeitschrift des Vereines Deutscher Ingenieure*":

After having pointed out the defects in the explanations offered at the official inquiry and carefully studied all the testimony; after having inspected the ship and the heaps of ruins at J. and G. Thomson's establishment, he comes to the following conclusions: When the ship was last docked, in January of 1890, it was found that while the bracket bearing of the port engine was in good condition, that of the starboard engine had worn down about $\frac{3}{16}$ inch. Instead of looking up the cause of such unusual wear, new lignum-vitæ strips were fitted and the ship put in service. When finally, after the accident had taken place, the ship was docked, and the after section of the starboard propeller shaft and propeller dropped to the bottom of the dock, it was discovered that the outboard coupling disc of the inboard section of propeller shaft had been partially broken off, a crack running around the flange at its junction with the shaft body. Portions of this crack showed rust when inspected shortly after the fall, which took place in the night. Mr. Mueller argues from this, and from the fact that the other flange remained intact, that the crack was an old one. If this is accepted, it appears at once that the shaft must have run untrue, in course of time wearing away the bearing, as it did, until the extra strain thrown on the shaft broke it and the engines ran away. He severely criticises the machinists and oilers of the steamer for not having noticed anything wrong in the working of the star-

board shaft, especially as they had the port shaft to make comparisons with, and shows that at the time of the accident there were only four men on watch in both engine rooms, of which number two belonged to another watch and were relieving the same man. The rest of a probable number of twelve were getting their tea.

Before going on with an explanation of what followed the breaking of the shaft, it is necessary to state that Mr. Mueller found the cast steel of the columns and the piston a brittle and unreliable material. He discovered this when he attempted to cut off chips. These always broke off short. Of the six columns of the starboard engine five had been patched. Mr. Mueller found that the thickness of material, not only in the cast-steel columns but also in the cylinders and valve chests, varied between very wide limits, the thickness of metal in the columns, intended to be one inch, varying from about $\frac{21}{32}$ and $\frac{3}{4}$ inch to $1\frac{1}{4}$ and even $1\frac{3}{4}$ inches. Such castings must of necessity be full of initial strains. The piston was lighter than ordinary practice demands. The packing rings were separated one inch, this space being occupied by a spring. The piston rod has a flange at its upper end 23 inches in diameter, $4\frac{1}{2}$ inches thick, and the piston is through bolted to it by eight $3\frac{1}{4}$-inch bolts.

The shaft being broken, the revolutions quickly increased to a point where the additional strains exerted upon the piston, due to its own inertia, were sufficient to fracture it; or else the spring between the packing rings broke first, and the cast-iron rings slamming soon broke up the rim. The whole piston was then speedily demolished, and the pieces, caught between the flange of the piston rod and the cylinder head and bottom, forced out both. Pieces falling down were caught between the upper end of the connecting rod and the crosshead guides, and while the lower end kept on rotating the upper end broke up the columns, pressing out portions with sufficient force to project them against the midship bulkhead and perforate it. A larger piece dropping finally led to the collapse of the columns, and the whole structure came down, the cylinder head dropping through and falling undermost. The connecting rod fell to

starboard, demolishing the condenser, the piston rod to port. The H.P. and I.P. cylinders kept on turning over a while longer, until the L.P. crank caught on the obstructions accumulating in the crank pit, the shaft was lifted up from its bearings, tearing the bulkhead separating the engine from the dynamo room, and the engine came to a standstill. The theory that the connecting rod caused the breaking up of the columns in the manner described is supported by the fact that it was found bent without showing an appreciable indentation on its concave side.

DELTA METAL.

The following data is extracted from the pages of Vol. 102 of the "Proceedings of the Institution of Civil Engineers," into which it was translated from *Annalen für Gewerbe und Banwesen* for 1890:

The process of manufacture of delta metal is as follows: When iron is heated in molten zinc, the latter dissolves about nine per cent. by weight of iron, the exact point of saturation depending upon the temperature at which the zinc is maintained during the process. In order to obtain uniform results, it is thus important to keep the zinc at a constant temperature, viz., the cherry-red heat of iron, at which temperature zinc absorbs 8.5 per cent. of iron. This mixture of zinc and iron is then cast into bars, and is then mixed with certain quantities of copper or copper and tin, according to the purpose for which it is to be used. A certain proportion of copper manganese is employed.

The delta metal is as soft as wrought iron, has the strength of steel, and possesses a golden yellow color. It can be forged at dull redness, drawn into wire, welded, and when cast runs freely and gives a fine grain on fracture.

It is not affected by the atmosphere, does not oxidize, and resists to an extraordinary degree the effects of acids and sea-water. The experiments of Prof. Unwin are quoted as indicating that at high temperatures this metal, both rolled and cast, is much stronger than phosphor-bronze, the average tensile strength being 83,619 pounds per square inch of area.

The metal melts at a temperature of 1742° Fahr. and has a specific gravity of 8.6.

The following table of tests demonstrates the force of the claims of superiority of delta metal:

Delta metal,	green sand casting, breaking strain.....	46,816 lbs. per square inch.
Do.	rolled hard (1½ inch bars).....	76,036 lbs. per square inch.
Do.	annealed.....	66,752 lbs. per square inch.
Do.	drawn into wire, 22 B. W. G.....	140,000 lbs. per square inch.

COAL IN TONQUIN.

Quite an important, as well as interesting bit of information comes to the Western world in the discovery and test of a semi-anthracite coal in Tonquin.

Mr. William Warren, who has recently returned to England from the East, reports as the result of extensive prospecting operations by the French in that newly acquired territory, the following data:

The seams of coal crop out over an extensive area among the hills fringing the Gulf of Tonquin. One of the seams is 152 feet thick. It is a coal of fine steaming quality, having about 87 per cent. of fixed carbon, is free from sulphur, possesses the notable characteristic of containing less than 3 per cent. of ash, and is quite as smokeless as pure anthracite. The coal has been tried on a steamer, three hundred tons of this coal having been consumed with very satisfactory results, the vessel steaming well at a fully maintained speed with almost the same consumption as with Cardiff coal.

This discovery will be a serious blow to the Japanese coal industry, Hong Kong alone taking some six hundred thousand tons per annum. The Japanese coals contain, as an average, rather more ash than the general run of bituminous coals, while the Tonquin variety seems to contain considerably less, rendering this latter more economical and efficient with equal weight.

Should this coal find its way to market, it would be well worth a full test of its quality for our vessels on the China station.

SHIPS.

UNITED STATES.

The President has designated the names for the following vessels under construction for the Government:

Cruiser No. 7, building at the Navy Yard, New York, will be called the *Cincinnati.*

Cruiser No. 8, at the Navy Yard, Norfolk, Va., is the *Raleigh.*

Coast-line battle-ship No. 1, under construction at Cramps', Philadelphia, will be the *Indiana.*

Coast-line battle-ship No. 2, also being built by the Cramps, is the *Massachusetts.*

Coast-line battle-ship No. 3, now under contract at the Union Iron Works, San Francisco, is named the *Oregon.*

The armored cruiser No. 2, of 8,150 tons displacement, building by the Cramps, is to be called the *New York,* this name being for the State and not for the city; the provisions of the act of Congress being to the effect that first-rates shall be named after States.

The *New York* is to be lengthened twelve feet, making her length over all 392 feet, and increasing her displacement from 8,100 tons to about 8,350.

On Dec. 20, bids were opened at the Navy Department for the *Harbor Defense Ram* described on page 576 of the last number of the JOURNAL. The only bid submitted was that of the Bath Iron Works, $930,000, on the Department's design of hull and machinery, and the contract was awarded to that firm.

On the same date bids were opened for a sea-going *Torpedo Boat,* advertised for Oct. 18, 1890. Displacement on trial to be 112 tons, about. Guaranteed speed in smooth water for two hours, 24 knots. For each quarter knot exceeding 24 knots up to and including 25 knots, a premium of $1,500 will be paid the contractors over and above the contract price. For every quarter knot in excess of 25 knots, the premium will be $2,000. If the boat fails to make 24 knots under the specified conditions, a

penalty of $5,000 will be enacted. Minimum speed necessary for acceptance, 23 knots.

Three bids were received; two from the Herreshoff Manufacturing Co., and one from the Cowles Engineering Co.

The bids of the Herreshoff Co. were $125,000, with $5,500 additional for galvanizing, and $93,400 with $4,000 additional for galvanizing. The bid of the Cowles Co. was $119,940. The final award of contract has not yet been made.

Bids for the *Torpedo Cruiser No. 1* were to be opened Feb. 11th. No bids were submitted, however, and it is doubtful whether the vessel will be built.

The design offered many novel features, and the following description will be of interest: Length on L. W. L., 259 feet; beam, extreme, 27.5 feet; mean draught, 9 feet; displacement on trial 800 tons (40 tons of coal on board); coal capacity, 175 tons.

There is a protective deck extending the full length of the vessel. Thickness on flat, ½ inch; on the slopes, ¾ inch.

A double bottom extends the full length of the machinery and ammunition spaces. Inside the double bottom the cruiser is built on the longitudinal bracket system, the transverse frames being spaced ten feet apart, the longitudinals 18 inches. These frames are continuous, of 10 lb. plates with lightening holes, except on water-tight divisions. The angle bars for floors and longitudinals are $3 \times 2\frac{1}{2} \times 5$ lbs. The longitudinal bars are stiffened at the transverse frames by double 6-lb. bracket plates, and are further stiffened at two points in each frame space by pieces of channel bar split and opened, and riveted to the inner and outer longitudinal angle bars. The inner keel and the outer longitudinals are continuous. Outside of the double bottom the transverse frames are spaced 30 inches apart; two stringers, the protective deck, the upper deck and the fore and aft coal bunker bulkheads at the vessel's sides providing longitudinal strength. Opposite each of the frames outside the double bottom, intermediate between those within the double bottom, are bracket plates, one at the outer longitudinal, one at the inner keel. The general weight of outer plating is 10 lbs. per square foot; that of garboard and shear strakes, 20 lbs. There are bulb iron bilge

keels. There are fourteen thwartship water-tight bulkheads below the protective deck, of which eleven are continued to the upper deck; and twelve water-tight compartments in the double bottom. The coal bunkers are fore and aft on each side, extending throughout the boiler and the greater part of the engine space; total length 90 feet. They extend upward above the protective deck, leaving only sufficient space below the upper deck for ventilating and lighting purposes. Patent fuel is stored in the bunkers above the protective deck, and cellulose is packed in the space formed by the slope of the protective deck and the ship's sides beyond the bunker space to one foot above L. W. L. The rudder has 80 square feet of surface. It is one third balanced, and is operated by oscillating hydraulic cylinders. Weights: Hull and fittings proper, 441.27 tons; coal (on trial), 40 tons; equipment, outfit and stores, 21.38 tons; boats, 5 tons; crew, provisions, and miscellaneous, 28.25 tons; electric light outfit, 10 tons; armament and ammunition, 44 tons; propelling machinery, 252.45 tons; total, 842.35 tons.

The propelling engines are of the vertical, direct-acting, triple-expansion type with four cylinders, placed in a common water-tight compartment. Cylinder diameters, 23¾, 34½, two of 38, by 18 inches stroke. Estimated I.H.P. of main engines and circulating pump engine about 6,000, main engines making 333 revolutions. The H.P. cylinders will be forward. The H.P. cylinder has one piston valve, and the I.P. and the L.P. cylinders each two. The I.P. cylinder is cast with its own, as well as with the H.P. valve chest. The L.P. cylinders form one casting. The valves are worked by Stephenson links, the L.P. valves being worked through intervention of a rock shaft. Each main piston has two piston rods secured to a crosshead having flat extensions working between guides supported from the columns. There are four of these for each cylinder, round, of forged steel, trussed by forged steel stays. The peculiar arrangement of crosshead and guides was selected as giving a minimum weight of reciprocating parts. The crank shafts are made in one section. The condenser is made entirely of composition and rolled brass, and will have a cooling surface of about 8,400 square feet.

There will be two vertical, single-acting air pumps for each engine, worked off the low-pressure crossheads. The circulating pumps are centrifugal, worked independently. There will be eight water-tube boilers in two water-tight compartments, constructed for a working pressure of 200 lbs. The forced-draft system will consist of two blowers in each fire room, discharging into the fire rooms, which may be made air tight.

ENGLAND.

The English naval authorities have at length formed the determination not to depend entirely upon private firms for the supply and manufacture of machinery for war ships.

Recently orders have been given for plans and specifications to be prepared for the engines of the *Forte*, a ship which is to be laid down in April. These engines are to be of the vertical, triple-expansion type, are to be designed for 9,000 I.H.P., and will be constructed at Chatham Dock Yard. The new vessel will have a displacement of 3,600 tons, and is to have her stern, stern-post and rudder-post of phosphor-bronze, and her bottom will be sheathed with wood and coppered.

The *Wallaroo*, formerly *Persian*, the last of three of this type, a part of the new Colonial fleet, has been completed at the Elswick works, and on December 13, 1890, she made her final trial trip of eight hours.

The propelling machinery consists of two sets of inverted vertical, triple-expansion engines in separate water-tight compartments, having a collective I.H.P. of 4,500 under natural draft, and 7,000 with forced draft. The cylinders are 31, 46 and 70 inches in diameter, with a common stroke of 33 inches.

Steam is supplied at a pressure of 155 pounds per square inch by four double-ended return-tube boilers, each boiler having four furnaces.

Throughout the eight-hour trial the engines and boilers performed satisfactorily.

The following data was collected as the result of the trial: Steam pressure at boilers, 145; vacuum, starboard condenser, 28.2 (?); vacuum, port condenser, 28.4 (?); revolutions of star-

board engine, 149; revolutions of port engine, 146.8; I. H. P. starboard, 2278; I. H. P. port, 2285; collective I. H. P., 4563; being 63 I. H. P. in excess of contract requirements.

Mean air pressure in fire rooms during the trial was .57 inch of water.

Pique.—On the 13th of December, 1890, there was launched from the yard of Sir C. M. Palmer's Shipbuilding and Iron Company, at Howden-on-Tyne, a fine addition to the fleet of the English Navy, the *Pique*, a second-class twin-screw protected cruiser. Her principal dimensions are: Length between perpendiculars, 300 feet; extreme breadth, 43.66 feet; moulded depth, 22.75 feet; mean draught, 17.5 feet; displacement at this draught, 3,600 tons; I. H. P., 9,000; speed, 20 knots.

The vessel has two smoke-pipes, and two pole masts.

The hull is built of steel throughout, and the bottom is sheathed with teak three and a half inches thick to two feet above the mean load line. The stern, stern post, rudder and shaft brackets are to be of phosphor bronze. A steel protective deck extends the full length of the ship. The transverse section of the protective deck is in the form of a flat arch, the crown of which rises one foot above the water-line in the center of the vessel, sloping at the sides to a point four feet below the load-line. This armor is one inch thick on the crown and two inches thick on the sloping part.

Vertical engines will be fitted, and to insure protection to the cylinders, which project above the protective deck, a belt of steel armor five inches thick with a teak backing of seven inches is fitted around the engine hatchways between the protective and upper decks. A double bottom extends the full length of the engine and boiler spaces. The coal bunkers are placed alongside the engines and boilers, and extend to the upper deck. Twin-screws will be actuated by two sets of triple-expansion engines, each set being in a separate compartment, the cylinders being 33.5, 49 and 74 inches in diameter, with a common stroke of 39 inches. The boilers are five in number. The closed fire room system of forced draft will be used. The bunker capacity will be 400 tons. The armament

consists of two 6-inch breech-loading guns, six 4.7-inch quick-firing guns, and four 5-barrelled Nordenfeldt guns.

Naiad.—On the 29th of November, 1890, the second-class cruiser *Naiad* was launched from the yard of the Naval Construction and Armaments Company, Barrow, she being one of twenty-nine provided for by the naval defence act of 1890. She is 300 feet long, and in all other respects is a ship exactly similar to the *Pique*, already noticed.

Blanche.—Although so recently commissioned, a serious defect has already come to light. The propeller shafts are protected by a sheathing of brass, which in the case of the *Blanche* is said to have been riveted on in several sections. These sections are each four feet in length, and it has been discovered that each joint of this sheathing had allowed the sea water to penetrate to the shaft, setting up corrosion. The sheathing has been removed, and quite extensive oxidation is seen to have taken place, which in the course of time would have resulted in serious weakening of the shaft. It is feared that a similar state of affairs will be found to exist on the *Blonde*, a sister ship of the *Blanche*, fitted, in detail, in a precisely similar manner.

Barfleur.—The keel of the *Barfleur* was laid at Chatham dock-yard in January, 1891. She is to be a second-class battleship of the following dimensions: Length between perpendiculars, 360.5 feet; extreme beam, 70 feet; displacement, 10,500 tons.

The motive power will consist of triple-expansion engines, intended for the development of 12,000 I.H.P., which will drive the ship eighteen knots an hour.

Her armament will comprise four 10 inch, 29-ton breech-loading guns in barbettes; ten 36-pounders and seventeen quick-firing guns; six tubes for the discharge of Whitehead torpedoes.

No little comment has been raised by the announcement that very serious defects have been discovered in the machinery of one of the largest first-class battle-ships of the English Navy, the *Anson*, of the *Admiral* class, and one of the vessels of the Channel Squadron. On returning to port from a brief cruise she was placed in dock for inspection and the usual overhauling

preceding an assignment for duty with the Channel Squadron on its cruise.

An examination of the propellers and other stern fittings on the starboard side developed the fact that the shafts were very defective, both as regards coupling bolts and the frame bearings. Two of the coupling bolts were broken while others were considerably worn, and the shaft was not in line.

On the port side also, the coupling bolts were much worn, but none were found broken. These serious defects could not have been discovered by the engineer officers of the ship, and had the vessel proceeded to sea in that condition it is almost certain that very serious consequences, even to the loss of the ship, might have ensued. The cost of this ship was no less than $3,790.800.

THEORETICAL AND PRACTICAL COAL ENDURANCE OF WAR SHIPS.

Economical coal consumption has become one of the most important items of consideration in the science of steam marine engineering, and in a naval war ship it is truly of vital importance, for in all naval wars of the future those ships possessing the greatest coal endurance, other qualities being equal, will be the most valuable.

Unfortunately theory and practice, as in many other directions, do not mutually support each other, and unless extensive and costly experimentation is carried out, the real coal endurance of a modern war ship is a question of unknown quantities.

As an illustration of the foregoing, witness the experience of the English battle ships *Howe* and *Camperdown*, during the manœuvers of August, 1890.

Each of these vessels is credited with a coal capacity of 1,200 tons, a quantity sufficient, according to calculation, to carry them 7,200 knots at an average speed of ten knots an hour.

Unfortunately, those figures fail when tried in the crucible of experiment, as do also deductions founded on results obtained on the official trials either over the measured mile or during the four-hours' run.

Under forced draft on her official trial the *Camperdown* realized a speed of 17.14 knots, developing 11,742 I.H.P. on a coal

consumption of 3.26 lbs. per hour per horse-power. With natural draft the speed made was 16.3 knots on 8,622 I.H.P. The coal consumption being 2.11 lbs. per I.H.P. per hour; and at a ten-knot speed, the coal consumption, on the foregoing basis of comparison, would fall to 1.85 lbs. per hour per I.H.P., the actual power required at this speed not exceeding 1,600, equivalent to a daily consumption of less than forty tons.

To show that this is only what might be called a "smooth water" deduction it remains but to record that, after steaming four thousand knots at a speed somewhat under eight knots an hour, it was found on her arrival in port that she had not sufficient coal in her bunkers to steam a thousand miles at a ten knot speed.

Taking the case of the *Howe*, it is found that she is not quite so prodigal in her coal expenditure as her sister ship, the *Camperdown*, although at sixteen knots an hour the figures show but little difference, so that under like conditions of trim and displacement the coal consumption at a speed of ten knots should be about the same. Under forced draft a speed of 16.932 knots was made with an I.H.P. of 11,703, the coal consumption being 2.17 pounds per hour per I.H.P.

Under natural draft the *Howe* made a speed of 15.872 knots, an I.H.P. of 7,695, consuming 1.99 pounds of coal per hour per indicated horse-power.

This vessel, after steaming a little over three thousand knots, at about ten knots an hour, ran short of coal in mid-ocean on the homeward voyage, and was forced to fall out of line to replenish her bunkers from the colliers following in the wake of Admiral Seymour's fleet.

Such performances are somewhat disappointing when compared with a promise of "ability to steam 7,250 miles at a ten-knot speed," and serve to emphasize the fact that tabulated figures for coal endurance deduced from trial trip results are evidently not to be treated as representing the ship's capability in actual practice.

The largest cruiser ever built at Devonport, England, was

launched from the Government dock-yard at that place November 24, 1890.

The vessel is the first-class, twin-screw steel cruiser *Edgar*, being one of nine ships provided for under the Naval Defence Act, and is the first to be launched. The following, of this class, are in various stages of construction at the following establishments:

The *Centaur* and *Crescent*, at Portsmouth; the *Grafton* and *Theus*, at the Thames Iron Works; the *Endymion* and *St. George*, at Earle's Shipbuilding Works, at Hull; the *Hawke* at Chatham, and the *Gibraltar* at R. Napier & Sons', Govan.

These vessels may be regarded as reduced copies of the *Blake* and *Blenheim*, the principal differences being in speed and coal capacity.

The *Edgar* is 360 feet between perpendiculars, and 384 feet over all; beam, 60 feet; draught, forward, 23.75 feet; aft, 24.5 feet; displacement at this draught, 7,350 tons. She is provided with a protective deck, affording protection to machinery, boilers, magazines, &c., varying in thickness from 2.5 to 5 inches.

The propelling machinery of this vessel has been constructed by the Fairfield Shipbuilding and Engine Company, Govan, and consists of two sets of triple-expansion engines, with cylinders of 40, 59 and 88 inches in diameter and a common stroke of 51 inches. Each engine will actuate a propeller of 16.5 feet in diameter and 23.33 feet pitch at 100 revolutions per minute, and the collective I.H.P. is intended to be 7,500 under natural draft and 12,000 under forced draft, the estimated speeds being 18 and 20 knots, respectively.

Steam will be supplied by four double-ended boilers, at a working pressure of 150 pounds per square inch. The heating surface is 20,000 square feet. The armament of the *Edgar* is unusually heavy, and will comprise two 9.2-inch, 22-ton B.L.R.; ten 6-inch quick-firing guns, and twelve 6-pounder quick-firing guns. The estimated cost, including machinery and armament, is $1,588,734.

FRANCE.

La Cecille.—Deck-protected cruiser. This vessel was built at La Seyne, near Toulon, and has been accepted by the French government.

The hull is of steel and of the following dimensions:

Length between pependiculars, 365.7 feet; greatest beam, 49.2 feet; depth of hold, 34.9 feet; mean draught, 19.78 feet; displacement, 5,766 tons.

This weight is distributed as follows:

Hull, including fittings, 1950 tons; armor, 750 tons; cellulose, 40 tons; boats, masts, rigging and various apparatus, 252 tons; crew, provisions and drinking water, 187 tons; engines, boilers and water in boilers, 1,225 tons; auxiliary machinery, 65 tons; coal, 700 tons; armament, ammunition and torpedo gear, 327 tons; miscellaneous, 63 tons; surplus, available for any purpose, 207 tons.

The armored deck, 1.6 feet above the water line in the center, is 1.6 inches thick in the center, and 3.2 inches thick at the inclined sides.

There are two athwartship bulkheads 3.2 inches thick, to protect the covered deck battery of ten 14-cm. guns. The bow along the water line and for about twenty-five feet each side is protected by 16.4 inches of armor. The cellulose is in a coffer-dam extending from 3.9 feet below to 4.5 feet above the water-line.

In addition to the ten 14-cm. guns, there are eight 16-cm. guns, fourteen 37-mm. Hotchkiss revolving guns, and four 47-mm. rapid-fire guns. The ship is provided with four torpedo tubes, one at the bow, one at the stern and two broadside. There are four sets of vertical, compound engines, having cylinders of 39.4 and 72.4 inches in diameter and 36-inch stroke. The twin screws are each four-bladed, and are made of manganeze bronze.

There are six double-ended boilers, three corrugated furnaces at each end, and intended for a working pressure of 96.3 pounds per square inch. Heating surface, 26,738 square feet; grate surface, 852.5 square feet.

For the trial trips the ship was fully laden, and in addition

there was placed on board one hundred tons of ballast, being the available surplus weight prescribed in the contract.

On the first trial, which was six hours in duration, the ship was run six times over the official course of 6.72 miles near the Hyères Islands.

The wind was fresh and sea moderate.

The mean results were as follows: Speed, 19.173 knots; I.H.P., 10,480; revolutions of engines, 101; steam, 89.68; air pressure in fire-room, 4.9 inches of water, and this pressure was easily maintained.

On the trial for maximum performance the conditions were more favorable, an I.H.P. of 10,680 being developed, giving the ship a speed of 19.436 knots.

The "coal-consumption" trial lasted twenty-four hours, the speed being maintained at 18 knots. During the trial the I.H.P. was 6,595 as a mean, and the coal per hour per I.H.P. was 1.98 pounds.

AUSTRIA.

The *Franz Josef I.* was built at the yards of S. Rocco, in the Stabilimento Tecino, Trieste, and was launched May 18, 1889. The ship is built of steel, and has a double bottom throughout the engine, boiler and ammunition rooms. She is provided with a protective deck, 2.16 inches in thickness, extending 4.1 feet below the water line. This deck, however, does not extend the whole length of the ship, being intended for the protection only of the most vital parts, engines, boilers, &c.

A protecting belt of cellulose extends from the termination of the protective deck, abreast of engines and boilers, to a safe distance toward the keel.

The twin-screws, of the Griffiths type, are of bronze, three-bladed, and are of 14.5 feet diameter and 20 feet pitch. Weight of engines, 890 tons. The bunker capacity is 660 tons of bituminous coal.

The trial trip under natural draft took place on the 8th of March of last year, 1890, and consisted of a four hours' back-and-forth run over a measured course of forty miles. At the be-

ginning of this trial the draught forward was 17.18 feet; aft, 20.3 feet, a mean of 18.74 feet, giving a displacement of 3,998 tons. Her immersed midship section is 762.5 square feet.

Through the efficiency of the ventilating apparatus the temperatures of the engine rooms varied between narrow limits, the minimum being 84.2° F. and the maximum 92.4° F.; fire rooms, 91.5° F. and 113.4° F.

The boiler pressure during the trial varied from 107 pounds per square inch above the atmosphere, as a minimum, to a maximum of 142 pounds.

The vacuum maintained in the condenser was upward of 25 inches of mercury.

Initial pressure in the three cylinders of one engine was H.P., 119; I.P., 41; L.P., 2.72, and the other engines a little less, viz., H.P., 113; I.P., 38; L.P., 1.9.

The total horse-power developed was 6,274, divided as follows: 3,173 for the first-mentioned engine and 3,101 for the second.

The mean number of revolutions for the trial was 103.5 per minute.

A mean speed of 17.84 knots was maintained, the requirements of the contract being 17 knots.

The forced-draft trial took place May 20, 1890, and consisted, like the former, of a four hours' run back and forth over a measured course of forty miles.

Draft at beginning of trial, forward, 17 feet, aft, 20.38 feet, a mean of 18.69 feet; displacement, 3,983 tons; immersed midship section 760.7 square feet.

Temperature on deck, 70° F.; forward engine room, 110° F.; after engine room, 88° F.; forward fire room, 110° F.; after fire room, 124° F.

The safety valves were set to blow off at 143 pounds, and the mean boiler pressure maintained, was 129 pounds.

A mean vacuum was maintained in the condenser of 25.2 inches of mercury.

Initial pressures in cylinders, H. P., 138.3; I. P., 52.6; L. P., 4.55, and H. P., 141; I. P., 51.2; L. P., 2.987.

Air pressure in the fire room varied from 2 to 3 inches of water.

Mean revolutions of engines, 111.77 and 111.8, developing a total horse-power of 8,305, divided as follows, 4,284 and 4,021.

The mean speed of the run *down* the course was 19.165 knots, while the speed up the course was but 18.625, a mean over the two courses of 18.895.

The slip of screws was 10.45 per cent. of their speed.

SPAIN.

Infanta Maria Teresa.—This vessel, one of the latest additions to the Spanish Navy list, is an armored cruiser constructed at the works of Martinez, Rivas & Palmer, of Bilboa, where two more cruisers of this type are in process of building.

The *Infanta Maria Teresa* was launched Sept. 30, 1890, has a hull built of mild steel of Spanish manufacture, and is divided into numerous water-tight compartments.

Her principal dimensions are as follows: Length over all, 360.75 feet; beam, 65.16 feet; mean draft, 21.87 feet; corresponding displacement, 7,000 tons.

The central side armor is 12.16 inches in thickness, and extends from 1.51 feet above the water line to a distance of 4.26 feet below. It was supplied by Cammell & Brown.

The protective deck consists of two courses of plates, in the middle part, increased to three courses at the sloping ends, forward and aft, each course being .9 inch thick.

The cylinders, which project through the protective deck, are enclosed by armor, 4.14 inches thick.

The motive power consists of two triple-expansion engines, in separate compartments, with cylinders of 37.8 inches, 61.85 inches, 92.13 inches in diameter, with a stroke of 45.6 inches.

The horse-power to be developed is expected to reach 9,000, giving a speed of 18 knots an hour under natural draft, and 13,000 I.H.P., with 20 knots under forced draft.

There are four doubled-ended and two single-ended boilers intended for a working pressure of 150.7 pounds. These boilers contain forty furnaces, each 38.98 inches in diameter. The total heating surface is 25,910 square feet.

SWEDEN.

[The following was contributed by a correspondent.]

Göta.—This is the second vessel built of a type between battle ship and cruiser, having armor and armament to conform in some degree with ships of the former class, whilst the displacement is more like that of cruisers in other navies. She is nearly in all respects similar to the *Swea*, which was accepted by the Government in Sept., 1886. *Göta* had a preliminary trial in Nov., last year, but owing to heating of one of the main pistons, it was discontinued and renewed in Jan., this year, when a perfect success was the result. Work on the *Göta* was commenced in Dec., 1887, at the Lindholmen S. & E. B. Co., domestic material being used all through, with exception of the heavy armor plates, which were taken from Creusot, in France. The displacement is 2,931 tons, the length being 265 feet 6 inches, by 49 feet 6 inches, beam, by 16 feet 6 inches draught when in fighting trim. Watertight subdivision has been carried out to the fullest extent, there being 210 such compartments, inclusive of those in the double bottom. A 12-inch armor belt, with the ordinary backing, runs a length of 149 feet, from amidship fore and aft. There is also a 2-inch protective steel deck over the machinery, stores and magazines, extending throughout the whole length. The battery consists of two 10-inch Armstrong breech-loading rifles, placed in one revolving turret protected by 12-inch armor, and has an angle of train of 300°; four 6-inch breech-loading rifles, of Bofors (Sweden) manufacture.

The secondary battery consists of four Nordenfelt four-barrel machine guns and six Gatling guns on the superstructure and mast. In the bow, right under the ram, is fitted a torpedo discharge tube; two similar tubes are also placed on the sides of the ship for Whitehead torpedoes. One military mast, which is hollow, extends from the conning tower, and carries two small machine guns. A complete electric plant is fitted for lighting purposes in general, and for the search lights, of which there are four. The twin-screw engines, which are horizontal, two-cylinder compound, developed 3,100 I.H.P., with a coal consumption of 2.45 lbs. of coal per I.H.P.

The main dimensions of machinery are as follows:

H.P. cylinder, 46 inches; L.P. cylinder, 80 inches; stroke, 32 inches; air pump, 28 inches diameter by 18½ inches stroke (single acting); circulating pump, 22 inches diameter by 18½ inches stroke (single acting); diameter of screw, 14 feet, pitch, 16 feet 6 inches; six single-ended boilers, carrying 72 pounds of steam.

At 93 to 94 revolutions a speed of 16 knots is expected.

Capacity of coal bunkers, 260 tons.

The ship can carry 60 days' provisions. Complement, 196.

ARGENTINE REPUBLIC.

The official steam trials of the last cruiser built for the Navy of the Argentine Republic, the *Veinte y Cinco de Mayo*, came off during the first part of November, 1890, in the presence of the Board sent to England for the purpose, and resulted most satisfactorily.

The ship was built by Armstrong, Mitchell & Co., and was designed by Philip Watts. She is 325 feet between perpendiculars, 43 feet beam, 16 feet mean draught, and her load displacement at that draught is 3,200 tons.

The contract required an I.H.P. of 8,500 under natural draft, and 13,500 with forced draft, the speeds required being 20.75 and 22 knots, respectively.

The machinery consists of two sets of four-cylinder, triple-expansion engines, working twin screws, and supplied with steam from four double-ended boilers. These engines were built by Humphreys, Tennant & Co., and lie wholly below the water-line, being further protected by a steel protective deck. In each set of engines there are two low-pressure cylinders of 66 inches diameter, one intermediate-pressure cylinder of 60 inches diameter, and one high-pressure cylinder of 38 inches diameter with a common stroke of 30 inches. The protective deck extends the whole length of the vessel, being 12 inches above the water line along its middle, the principal hatchways being protected by 5-inch armor glacis plates and coffer-dams. On the sloping sides of the protective deck the plating is from 3.5 to 4.5

inches in thickness, the horizontal parts being 1.75 inch thick. Coal bunkers are fitted above the protective deck along the sides of the ship minutely subdivided by water-tight bulkheads.

The normal coal supply is 300 tons, but bunker capacity for twice that quantity has been provided. The steam trials were made off the mouth of the Tyne, on the admiralty measured mile, a performance curve in terms of revolutions was constructed, and the mean speed determined by taking the total number of revolutions of the screws, obtaining therefrom the mean per minute, and applying this to the performance curve previously obtained. The trial was a run of six hours' duration under natural draft, and the actual performance, in speed, was 21.237 knots per hour. The mean I.H.P. developed during the six hours was 8,700 with a mean speed of engines of 144.9 revolulutions per minute.

So little importance was attached to the forced-draft trial by the Argentine Board, that the ship would have been accepted without it, but in spite of this the trial was made. After a day's steaming, runs over the measured mile were made under forced draft, and the result was a mean speed of 22.43 knots per hour with 160 revolutions per minute, and an I.H.P. of 13,800.

The general designs of the *Veinte y Cinco de Mayo* are similar to those of the *Piemonte*, completed by the same firm in 1889 for the Italian Government, save that the former is of some 700 tons greater displacement than the latter. The *Piemonte's* best performance was 20.4 knots as compared with 21.237 obtained by the *Veinte y Cinco de Mayo.*

In this connection it will be interesting to note some of the foreign high-speed cruisers now in course of construction.

Government.	Name of ship.	Displacement.	Length.	Speed.		I. H. P.	
				Forced draft.	Natural draft.	Forced draft.	Natural draft.
British	*Blake, Blenheim* ...	9.000	375.0	22.0	20.0	20.000	...
Do	*Vulcan* and...	6,620	350 0	20.0	...	12,000	...
Do	*Edgar* class	7,350	360.0	20.0	...	12,000	...
Spanish	*Almirante Quendo*..	7,000	365.0	20 0	...	15,000	...
Do	*Alfonso XIII* and *Lepanto.*	5,000	318.5	20.0	...	11,000	...
French.....	*Alger, Isly* and *Jean Bart.*	4.160	346 0	19 0	...	8,000	...
Do	*Dupuy de Lôme*.....	6,297	376.0	20.0	...	14.000	...
Do	*Charner*.....	4.745	348.0	20.0	...	8 300	...
Do	*Davoust*.....	3.027	297.5	20 0	...	9.000	...
Italian.....	*Liguria* class	2,280	262.5	20.0	...	6,500	...

The latest, and in some respects the most important addition to the Argentine Navy, was subjected to a trial on the 13th of last November.

This was a Yarrow torpedo boat with quadruple-expansion engines. The boat in question is the last of fourteen built for that Navy. The *Bathurst*, for this is the name given to the boat under discussion, is 130 feet long, and 13.5 feet beam on the water line.

The skin plating, which is of steel and galvanized, is of greater thickness than is usual with this class of vessels, and there are no less than eleven water-tight compartments. Forward there is a high turtle back terminating in a conning tower.

The boilers are of the locomotive-marine type usually fitted by this firm, having copper fire boxes and brass tubes, and, as in previous examples of this method of construction, no trouble whatever was experienced, either at high or low pressures, from leaky tubes.

In the *Bathurst* the chief point of interest is the motive machinery. All told, there are in the engine compartment of 20 feet in length no less than six auxiliary engines besides the main engines with four cylinders and the condenser.

This craft has a very good arrangement for maintaining a vacuum in the condenser when the main engine and air pump is at rest. This arrangement is a small auxiliary air pump attached to the circulating pump, and as it is always running it is able to keep the condenser free from water and maintain a moderate vacuum when steam is not coming from the main engines. This arrangement adds considerably to the manœuvering capacity of the boat and works very well in practice.

One of the main objects in fitting the *Bathurst* with quadruple expansion engines was to reduce as far as possible the unpleasant, if not dangerous, vibrations so prominent a feature in many of the high-powered short-stroke engines, and the results of the trials with this boat have demonstrated the wisdom of the arrangement. The plan adopted by the Yarrow Company to obviate this vibration has been to arrange the four cylinders in pairs, each pair having its cranks in such a position that the reciprocating parts balance each other.

In addition to decreasing vibration by the introduction of the fourth cylinder, there has been an increase in the horse-power developed, so that while the triple-expansion engines of the other boats of this class developed 1120, the *Bathurst* has made 1230 horse-power, a clear gain in horse-power of 110. This additional I.H.P. has increased the speed of this boat rather more than a knot as compared with the five others of the same class fitted with triple-expansion engines.

The extra weight imposed by the additional cylinder is about three tons, and it is estimated that forty additional horse-power would be required to drive this three tons of extra weight, in order to get the same speed as the triple-expansion-engine boats. That is to say, if the *Bathurst's* engines weighed three tons more than those of the others, and only indicated forty horse-power more, the speeds of both types would have been alike, viz: 23.3 knots. But by these trials it is seen rather more than a knot in excess was obtained, so it may be said, approximately, that there is a clear gain of 70 I.H.P.

The base of the engine is formed of longitudinal angle steel with deep vertical flanges, and is attached to the main structure

of the vessel throughout the length of the engine room. These longitudinals are connected by cross-pieces formed by steel castings, the whole forming a foundation upon which are erected the steel columns of the engine framing.

The oil service for lubrication is exceedingly efficient, each bearing having its own oil duct, controlled by a small valve, the supply oil being at all times visible. There are seven collars in the thrust bearing, and as there is a sliding coupling between the crank shaft and thrust shaft, the wear of the collars is taken up automatically and adjustment is unnecessary.

Below will be seen the official data of the trial.

"OFFICIAL TRIAL IN THE "LOWER HOPE," RIVER THAMES, OF A SEAGOING TORPEDO BOAT WITH QUADRUPLE-EXPANSION ENGINES BY YARROW & CO., FOR THE ARGENTINE GOVERNMENT. LOAD CARRIED, 12 TONS. DISPLACEMENT, 75.5 TONS."

Steam in boiler.	First receiver.	Second receiver.	Third receiver.	Vacuum in inches.	Air pressure in stokehold.	Revolutions on knot.	Revolutions per minute.	Time on knot.	Speed.	Mean Speed.	Second mean.	True mean.
lb.	*lb.*	*lb.*	*lb.*		*in.*			*m. s.*	*knots.*	*knots.*	*knots.*	*knots.*
199	74	35	4	23.7	3.3	1080	432.	2 30	24.000			
										24.587		
200	74	35	4	23.5	3.0	1030	432.	2 23	25.174		24.469	
										24.351		
200	75	35	4	23.5	3.2	1100	431.4	2 33	23.529		24.395	
										24.440		24.453
200	74	35	4	23.2	3.0	1020	431.	2 22	25.352		24.402	
										24.364		
200	74	35	4	23.0	3.4	1120	436.3	2 34	23.376		24.547	
										24.731		
200	74	35	4	23.2	3.0	1010	439.	2 18	26.086			

The total number of revolutions made in two hours at full speed with 200 lbs. in boiler were 51,848. The mean revolutions to make a knot by the formula $\frac{(t \times r_1) + (t_1 \times r)}{t + t_1}$ is 1061.3, and the mean speed by revolutions for the two hours 24.426 knots. The consumption of fuel per hour was 25¼ cwt.

November 13, 1890.

JAPAN.

The Japanese protected cruiser *Itsukusima*, built by the Societé des Forges et Chantiers, completed her contractors' trials recently with the following results:

At full load displacement with natural draft her speed was 15.72 knots per hour, while under forced draft a speed of 17.78 was maintained.

The following data is compiled from "Mitheilungen aus dem Gebiete des Seewesens":

Length between perpendiculars, 324.9 feet; maximum beam, 50.9 feet; depth of hold, 34.7 feet; mean draught, 19.8 feet; displacement at this draught, 4,277 tons; weight of armor, 200 tons; weight of armament, 475 tons.

The armament consists of one 12.6-inch 65-ton Canet gun in a turret forward; eleven 4.7-inch Canet rapid-fire guns, ten in broadside, one astern; eleven 47-mm. revolving cannon.

Thickness of turret armor, 11.8 inches; of barbette protecting base of turret, 4 inches; of protective deck, 1.6 inches.

MERCHANT STEAMERS.

S.S. Iroquois.—The following data will be of interest, as showing the success met with in designing and building a ship to meet extraordinary conditions.

The *S. S. Iroquois*, of the Jacksonville Line, was designed to run on the St. John's River, Fla., and the limit of draught was ten feet.

Her principal dimensions are:

HULL.

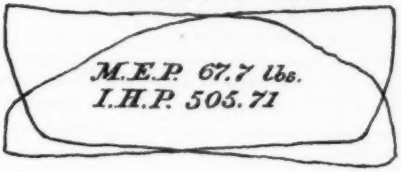
Length between perpendiculars	280 feet.
Beam, molded	46 feet.
Draught with 1,000 tons of dead-weight cargo	10 feet.
Depth from base-line to top of main deck beams at side	20.5 feet.
Main deck to upper deck	8 feet.

The *Iroquois* is furnished with four return-tubular boilers, 11 feet diameter and 12 feet long.

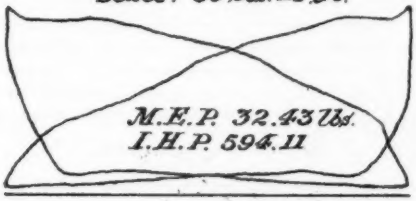
The engine is of the triple-expansion type, the cylinders having diameters of H.P., 23 inches; I.P., 36 inches; L.P., 60 inches, with a common stroke of 36 inches; boiler pressure, 160 pounds heating surface, 1,480 square feet; grate surface, 184 square feet; diameter of propeller, 13 feet; mean pitch, 15 feet.

Mean draught	12 feet 8 inches.
Displacement	2,890 tons.
Hours in motion	141.
Hours favorable winds	28.
Hours adverse winds	45.
Knots per voyage	1,660.
Speed in knots per hour	11.77
Total revolutions, voyage	746,245.
Revolutions per minute	88.2
Steam in boilers	149.2
Steam in 1st receiver	55.1
Steam in 2d receiver	4.8
Vacuum	26.9
Cut-off { 1st cylinder	26.4
Cut-off { 2d cylinder	19.8
Cut-off { 3d cylinder	22.5

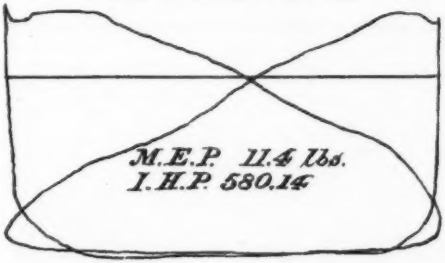
H.P. Cyl.
Scale: 100 lbs. = 1 in.



I. P. Cyl.
Scale: 50 lbs. = 1 in.



L.P. Cyl.
Scale: 12 lbs. = 1 in.



Total I.H.P. 1679.96

Boiler Pressure 160 lbs.

Press. in 1st Recr. 60 lbs.

do. do. 2d do. 9 lbs.

Vacuum ------- 27 ins.

Rev's. ---------- 99

Throttle		.975
Temperatures	Sea	45.
	Discharge	85.8
	Feed from condenser	112.55
	Feed from heater	139.
	Engine room	78.8
Indicated horse-power		1,625.94
Coal consumed per voyage		210 tons.
Coal consumed per day		37.5 tons.
Coal consumed per hour		3,407 pounds.
Coal consumed per I.H.P		2.09 pounds.
Initial pressure		157.5
Revolutions per minute		93.58
Vacuum		27.
Mean pressures		68.51 — 33.85 — 11.55
I.H.P		484.27 — 586.35 — 555.32
Total I.H.P		1,625.94
Coal per square foot of grate		19.6 pounds.
Average consumption of coal for twenty-seven (27) voyages		190.8 tons.

HIGHEST POWER OBTAINED.

Steam	160 pounds.
Revolutions	105.
Vacuum	27 inches.
Mean pressures	65.62 — 33.46 — 12.6
I.H.P	520.36 — 646.78 — 679.14
Total I.H.P	1,846.28

NOTE.—The horse-power given for average working is the mean of twelve (12) sets of cards.

The ANNUAL MEETING for the election of officers and transaction of other business took place at the Navy Department, January 15, Passed Assistant Engineer G. W. Baird in the chair.

The vote resulted in the election of Passed Assistant Engineer G. W. Baird as President, Assistant Engineer E. Theiss as Secretary-Treasurer, Chief Engineer David Smith, and Assistant Engineers S. H. Leonard and W. M. McFarland as Members of Council. Mr. McFarland declined a renomination for Secretary, owing to a press of private affairs.

From his report, which has since been made, it appears that the Society is in a flourishing condition. The membership has grown to three hundred, and the accounts show a surplus of four hundred dollars in the Treasurer's hands at the end of 1890.

A few remarks followed in continuation of the discussion on "Trial Trips", begun at the December meeting, whereupon adjournment took place.

In this connection and in view of the fact that the publication of the discussion on "Trial Trips and the Lessons to be Learned from Them," has been put off until May, it would be well to give the concluding remarks of Chief Engineer N. P. Towne in a paper read at the December meeting :

"In closing the second year of our Society's existence, I think we can congratulate ourselves on the position we have attained in the engineering world as a society.

"Our journal has steadily increased in circulation and interest, and we are steadily gaining members from the most eminent American and foreign engineers.

"I beg to thank, in the name of our Society, the members who have contributed to the valuable and instructive articles which we have published, and also to acknowledge our deep obligations to Messrs. Griffin and McFarland, our secretaries and editors, who have given so many of their spare moments and so much of their marked abilities to the good of our Society and profession."

LIST OF MEMBERS OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

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Hon. J. R. Soley, Assistant Secretary of the Navy.
Engineer-in-Chief G. W. Melville, U. S. Navy.

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Addicks, W. R., Superintendent Boston Gaslight Co.....24 West 4th street, Boston.
Allderdice, W. H., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Allen, F. B., Vice-President Hartford Steam Boiler Inspection and Insurance Co...Hartford, Conn.
American Society of Mechanical Engineers.....12 West 31st street, New York.
American Shipbuilder.....115 Broad street, New York.
Anderson, M. A., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Andrade, C., Chief Engineer, U. S. N.....U. S. S. *Yorktown.*
Army and Navy Journal.....240 Broadway, New York.
Aston, Ralph, Chief Engineer, U. S. N.....Delaware River Iron Works, Chester, Pa.
Ayres, S. L. P., Chief Engineer, U. S. N.....Member Board of Examiners, Phila., Pa.
Bailey, W. H., Agent American Tube Works.....20 Gold street, New York.
Bailey, F. H., Passed Assistant Engineer, U. S. N.....U. S. S. *Chicago.*
Baird, G. W., Passed Assistant Engineer, U. S. N.....
Assistant Supt. State, War and Navy Dept. Bldg., Washington, D. C.
Baker, C. H., Chief Engineer, U. S. N.....Navy Yard, New York.
Barber, F. M., Commander, U. S N.....The "Portland," Washington, D. C.
Barnard, G. A., Steam Power Plants and Steam Heating Apparatus, 15 Cortlandt street, New York.
Barry, J. J., Passed Assistant Engineer, U. S. N.....107 Henry street, Brooklyn.
Bartlett, F. W., Passed Assistant Engineer, U. S. N.....U. S. S. *Charleston.*
Barton, J. K., Passed Assistant Engineer, U. S. N.....U. S. Naval Academy, Annapolis, Md.
Bates, A. B., Passed Assistant Engineer, U. S. N.....U. S. S. *Minnesota.*
Bath Iron Works.....Bath, Me.
Baxter, W. J., Assistant Naval Constructor.....Navy Yard, Norfolk, Va.
Bayley, W. B., Passed Assistant Engineer, U. S. N.....U. S. Fish Commission, Washington, D. C.
Beavor-Webb J., Naval Architect.....45 Broadway, New York.
Bennett, F. M., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Bevington, M., Assistant Engineer, U. S. N.....Homestead Steel Works, Munhall, Pa.
Beach, E. L., Assistant Engineer, U. S. N.....U. S. S. *Philadelphia.*
Bieg, F. C., Passed Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Blake Manufacturing Company.....East Cambridge, Mass.
Blomberg, C. A.,.....Wm. Cramp & Sons S. and E. B. Co., Philadelphia, Pa.
Blue, Victor, Assistant Engineer, U. S. N.....U. S. S. *Pensacola.*
Bolten, A.,.....117 Market street, San Francisco, Cal.
Bonneville, A. A.....77 Liberty street, New York.
Borthwick, J. L. D., Chief Engineer, U. S. N.....Erie, Pa.
Bowers, F. C., Assistant Engineer, U. S. N.....U. S. Fish Commission Steamer *Fishhawk.*
Boyd, J. S., Superintending Engineer Atlantic Works.....East Boston, Mass.
Brecht, T. C.....Bureau Steam Engineering, Navy Department, Washington, D. C.
Brereton, J. I., First Assistant Engineer, U. S. R. M.....Cutter *Woodbury*, Rockland, Me.
Brooks, W. B, Chief Engineer, U. S. N.....Richmond Locomotive Works, Richmond, Va.
Brown, Jefferson, Chief Engineer, U. S. N.....Navy Yard, New York.

Bryan, B. C., Assistant Engineer, U. S. N.....U. S. S. *Cushing*.
Buehler, W. G., Chief Engineer, U. S. N.....124 South 17th street, Philadelphia.
Burd, G. E., Assistant Engineer, U. S. N.....U. S. S. *Atlanta*.
Bureau Steam Engineering Library.....Navy Department, Washington, D. C.
Burgdorff, T. F., Passed Assistant Engineer, U. S. N...University of Tennessee, Knoxville, Tenn.
Burgess, Edward, Naval Architect.....50 State street, Boston.
Butterworth, J., Chief Engineer, U. S. N.....25 William street, Cambridgeport, Mass.
Canaga, A. B., Passed Assistant Engineer, U. S. N.....Cornell University, Ithaca, N. Y.
Carr, C. A., Assistant Engineer, U S. N.....Navy Department, Washington, D. C.
Carter, T. F., Assistant Engineer, U. S. N.....U. S. C. S. S. *Patterson*.
Cathcart, W. L., Consulting Engineer.....Hoyt, Montgomery County, Pa.
Chalker, J. H., First Assistant Engineer, U. S. R. M.....U. S. R. Steamer *Seward*, Mobile, Ala.
Chambers, W. H., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Cleaver, H. T., Passed Assistant Engineer, U. S. N.....U. S. S. *Palos*.
Conant, F. H., Assistant Engineer, U. S. N.....Navy Yard, New York.
Cooley, M. E., Professor Mechanical Engineering.....University of Michigan, Ann Arbor, Mich.
Coryell, Miers, Consulting Engineer.....21 East 21st street, N. Y.
Cowles, Wm., President Cowles Engineering Company.....Foot 43d street, South Brooklyn.
Cox, Irving, Marine Engineer.....50 Wall street, New York.
Cramp, E. S., Superintending Engineer.....Wm. Cramp & Sons S. and E. B. Company, Phila.
Cramp, B. H., B. H. Cramp & Co., Brass Founders.....Philadelphia.
Cramp, C. D., with B. H. Cramp & Co., Brass Founders.....Philadelphia.
Crawford, Robert, Passed Assistant Engineer, U. S. N.....328 Front street, Media, Pa.
Creighton, W. H. P., Assistant Engineer, U. S. N .....Purdue University, Lafayette, Ind.
Cust, Leopold.....3 Park Terrace, Govan, Glasgow, Scotland.
Davidson, M. T., Proprietor Davidson Steam Pump Works.....49 Keap street, Brooklyn.
Davis, L. D , Builder of Steam Launches.....Humboldt Iron Works, Erie, Pa.
Day, W. B., Assistant Engineer, U. S. N.....Union Iron Mills, 33d street, Pittsburgh, Pa.
Denig, R. G., Passed Assistant Engineer, U. S. N.....U. S. Naval Academy, Annapolis, Md.
De Valin, C. E., Chief Engineer, U. S. N.....1308 New Hampshire avenue, Washington, D. C.
Dickinson, R. T., Naval Architect.....Delaware River Iron Works, Chester, Pa.
Dowst, F. B.....with B. F. Sturtevant, Jamaica Plain, Mass.
Dripps, W. A., Consulting Engineer.....3324 Walnut street, Phila.
Dungan, W. W., Chief Engineer, U. S. N.....Navy Yard, New York.
Durand, W. F., Professor Mechanical Engineering.....Agricultural College, Ingham county, Mich.
Dyson, C. W., Assistant Engineer, U. S. N.....Navy Yard, Boston, Mass.
Eckart, W. R., Consulting Engineer.....217 Sansome street, San Francisco, Cal.
Ecole du Génie Maritime, Directeur de l'.....27 Quai de la Tournelle, Paris.
Edson, J. B., Recording Pressure Gauges .....87 Liberty street, New York.
Edwards, J. R., Passed Assistant Engineer, U. S. N., University of South Carolina, Columbia, S. C.
Eldridge, F. H., Passed Assistant Engineer, U. S. N.....Navy Yard, New York.
Emanuel, J. M., Passed Assistant Engineer, U. S. N.....1812 North 12th street, Philadelphia.
Emery, C. E., Dr., Consulting Engineer.....22 Cortlandt street, New York.
Engard, A. C., Passed Assistant Engineer, U. S. N.....U. S. S. *Newark*.
Farmer, Edward, Chief Engineer, U. S. N.....U. S. S. *Chicago*.
Ferguson, G. R., M. E. and Pat. Attorney.....26 South Grove street, East Orange, N. J.
Field, C. J., Field Engineering Company.....15 Cortlandt street, New York.
Fisher, Clark, Fisher & Norris.....Trenton, N. J.
Fitch, H. W., Chief Engineer, U. S. N.....U. S. Naval Academy, Annapolis, Md.
Fitts, J. H., Professor of Engineering.....Agricultural College, Blacksburg, Va.
Flach, Emil.....Bureau of Steam Engineering, Navy Department, Washington, D. C.
Fletcher, Andrew, Jr., W. & A. Fletcher.....157 West 73d street, New York.
Fletcher, W. H., W. & A. Fletcher.....12th and Hudson streets, Hoboken, N. J.
Ford, J. D., Passed Assistant Engineer, U. S. N.....U. S. S. *Alert*.
Forsyth, Robert, Engineer.....Union Iron Works, San Francisco, Cal.
Frear, H. P., Naval Architect.....Union Iron Works, San Francisco, Cal.
Freeman, E. R., Passed Assistant Engineer, U. S. N.....U. S. S. *Concord*.
French, D. M., First Assistant Engineer, U. S. R. M.....68 Royal street, New Orleans, La.

Gage, Howard, Passed Ass't Engineer, U. S. N., Wm. Cramp & Sons S. & E. B. Co...Philadelphia.
Gow, J. L., Passed Assistant Engineer, U. S. N.....U. S. Naval Academy, Annapolis, Md.
Gowen, F. H .....West Newbury, Essex County, Mass.
Grafly, W. D.....York and Almond streets, Philadelphia.
Greene, A. S., Chief Engineer, U. S. N .....Hot Springs, Ark.
Greene, D. M., Director Renssalaer Polytechnic Institute.....Troy, N. Y.
Greene, I. C., Hope Brass Works.....P. O. Box 272, Baltimore, Md.
Greene, L. R., Walworth Manufacturing Company.....Cambridge, Mass.
Griffin, R. S., Passed Assistant Engineer, U. S. N.....U. S. S. *Philadelphia.*
Gsantner, O. C., First Assistant Examiner.....Room 124, Patent Office, Washington, D. C.
Habighurst, C. J., Passed Assistant Engineer, U. S. N.....U. S. S. *Enterprise.*
Haines, Hiram, Mining Engineer .....Cordova, Walker County, Alabama.
Hall, H., Assistant Engineer, U. S. N.....U. S. S. *Alliance.*
Hall, R. T., Passed Assistant Engineer, U. S. N.....U. S. S. *Pensacola.*
Halstead, A. S., Assistant Engineer, U. S. N .....Homestead Steel Works, Munhall, Pa.
Handren, John,.....Handren & Robins, 126 Washington street, New York.
Hanford, F., Lieutenant Commander, U. S. N.....U. S. S. *Pensacola.*
Hannum, J. L., Passed Assistant Engineer, U. S. N.....U. S. S. *Iroquois.*
Hanscom, C.....Col. Iron Works & D. D. Co., Baltimore, Md.
Harris, W. H., Chief Engineer, U. S. N.....U. S. S. *Atlanta.*
Hartrath, Armin, Assistant Engineer, U. S. N.....U. S. S. *Enterprise.*
Hasson, W. F. C., Assistant Engineer, U. S. N.....Johns Hopkins University, Baltimore, Md.
Hayes, C. H., Assistant Engineer, U. S. N.....U. S. S. *Yorktown.*
Heaton, W. W., Chief Engineer, U. S. N...Quintard Iron Works, 12th street and avenue D, N. Y.
Henderson, A., Chief Engineer, U. S. N.....Navy Yard, Boston.
Henderson, A. P.....Box 398, Newport News, Va.
Henderson, Eliot M.....Seattle, Washington.
Henthorn, J. T., Remington & Henthorn,.....146 Westminster street, Providence, R. I.
Herbert, W. C., Assistant Engineer, U. S. N.....Wm. Cramp & Sons S. & E. B. Co., Philadelphia.
Herwig, H., Passed Assistant Engineer, U. S. N.....Navy Yard, Washington.
Hibbs, F. W., Assistant Engineer, U. S. N.....University of Glasgow, Glasgow, Scotland.
Higgins, R. B., Assistant Engineer, U. S. N.....U. S. S. *San Francisco.*
Hine, R. B., Chief Engineer, U. S. N.....U. S. S. *Concord.*
Hogan, T. J., Westinghouse Air Brake Co.....P. O. box 1034, Pittsburgh, Pa.
Hollis, I. N., Passed Assistant Engineer, U. S. N.....U. S. S. *Charleston.*
Howell, C. P., Passed Assistant Engineer, U. S. N.....U. S. S. *Baltimore.*
Howell, W. N.....Wm. Cramp & Sons S. & E. B. Co., Philadelphia.
Hunt, A. M., Assistant Engineer, U. S. N .....472 Broad street, Newark, N. J.
Hyde, Chas., Superintending Engineer Bath Iron Works.....Bath, Me.
Hyde, John S.....Bath Iron Works, Bath, Me.
Inch, Richard, Passed Assistant Engineer, U. S. N.....Navy Yard, Portsmouth, N. H.
Inch, Phil., Chief Engineer, U. S. N.....U. S. S. *Charleston.*
Institution of Civil Engineers.....25 Great George street, Westminster, London, S. W.
Isherwood, B. F., Chief Engineer, U. S. N.....111 East 36th street, New York.
Janson, E. N.....Bureau of Steam Engineering, Navy Department, Washington, D. C.
Jefferis, J. E., Chief Engineer, U. S. R. M.....U. S. R. S. *Corwin*, San Francisco, Cal.
Johnson, G. R., Chief Engineer, U. S. N.....1233 New Hampshire avenue, Washington, D. C.
Jones, D. P., Chief Engineer, U. S. N.....U. S. S. *Richmond*, Newport, R. I.
Jordan, L. S .....Wm. Cramp & Sons S. & E. B. Co., Philadelphia.
Kaemmerling, G., Assistant Engineer, U. S. N.....U. S. S. *Boston.*
Kafer, J. C., Passed Assistant Engineer, U. S. N.....Morgan Iron Works, 9th and E. R., New York.
Kearney, G. H., Passed Assistant Engineer, U. S. N .....Navy Yard, New York.
Keilholtz, Pierre O., Supt. U. S. Electric Light Co.....1829 Linden street, Baltimore, Md.
King, C. A. E., Assistant Engineer, U. S. N.....Col. Iron Works and D. D. Co., Baltimore, Md.
King, F. B., Superintendent Marine Department, Penn. Steel Works.....Sparrows Point, Md.
King, W. R., Passed Assistant Engineer, U. S. N .....Asheville, N. C.
Kinkaid, T. W., Assistant Engineer, U. S. N.....U. S. S. *Pinta.*
Kirby, A., Chief Engineer, U. S. N .....405 C street southeast, Washington, D. C.

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Kirby, F. E., Marine Engineer, Detroit Drydock Co.....Detroit, Mich.
Kloppenburg, H.....1009 Magnolia avenue, Elizabeth, N. J.
Knowles Steam Pump Co.....Warren, Mass.
Koester, O. W., Assistant Engineer, U. S. N.....U. S. S. *Essex.*
Kuehl, W. H., Bookseller.....73 Jaeger street, Berlin, W.
Kunstadter, J. J.....31 Broadway, New York.
Kutz, G. F., Chief Engineer, U. S. N.....Union Iron Works, San Francisco, Cal.
Laws, E., Chief Engineer, U. S. N.....U. S. S. *Pensacola.*
Leavitt, E. D., Consulting Engineer.....2 Central Square, Cambridgeport, Mass.
Leitch, R. R., Passed Assistant Engineer, U. S. N.....U. S. S. *Chicago.*
Leonard, J. C., Assistant Engineer, U. S. N.....U. S. C. S. Steamer *McArthur.*
Leonard, S. H., Jr., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Leopold, H. G., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Lincoln, J. N., Clyde Steamship Co.....5 Bowling Green, New York.
Little, W. N., Passed Assistant Engineer, U. S. N.....Navy Yard, New York.
Loring, C. H., Chief Engineer, U. S. N.....20 Bullfinch street, Boston.
Loring, H., Jr.....City Point Works, South Boston.
Lowe, J., Chief Engineer, U. S. N.....Navy Yard, Washington, D. C.
Lundborg, C. G., Naval Architect.....430 West 14th street, New York.
Maccarty, G. M. L., Chief Engineer, U. S. N.....U. S. S. *Omaha.*
Macomb, D. B., Chief Engineer, U. S. N.....34 Chestnut street, Charlestown, Mass.
Magee, E. A., Chief Engineer, U. S. N.....
Board of Inspectors of Merchant Steamships, Postoffice Building, New York.
Magee, G. W., Chief Engineer, U. S. N.....Navy Yard, New York.
Main, H., Passed Assistant Engineer, U. S. N.....U. S. S. *Boston.*
Manning, C. H., Passed Assistant Engineer, U. S. N...Supt. Asmoskeag Mills, Manchester, N. H.
Manning, Maxwell & Moore.....111 Liberty street, New York.
Marine Journal.....132 Nassau street, New York.
Maschmeyer, A. M. P.....Bureau Steam Engineering, Navy Dept., Washington, D. C.
Matthews, C. H., Assistant Engineer, U. S. N.....Bolton House, Harrisburg, Pa.
Matthews, E. M.....Wm. Cramp & Sons S. & E. B. Co., Phila.
Mattice, A. M., Consulting Engineer.....2 Central Square, Cambridgeport, Mass.
McAllister, A., Assistant Engineer, U. S. N.....Navy Yard, New York.
McAllister, C. A.....Bureau Steam Engineering, Navy Department, Washington, D. C.
McAlpine, K., Assistant Engineer, U. S. N.....Navy Yard, Norfolk, Va.
McCartney, D. P., Chief Engineer, U. S. N.....S. L. Moore & Sons, Elizabethport, N. J.
McCutcheon, J. F., Chief Engineer City Hall.....City Hall, Phila.
McElmell, J., Chief Engineer, U. S. N.....Member Board of Examiners, Phila.
McElmell, Thomas, Agent Standard Oil Co.....1931 Spring Garden street, Phila.
McFlroy, G. W., Assistant Engineer, U. S. N.....U. S. S. *San Francisco.*
McEwan, H. D., Chief Engineer, U. S. N.....U. S. S. *Boston.*
McFarland, W. M., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
McKean, F. G., Chief Engineer, U. S. N.....Navy Department, Washington, D. C.
McKean, J. S., Assistant Engineer, U. S. N.....U. S. S. *Chicago.*
McMakin, Jos., Suptdg. Engineer Columbia I. W. & D. D. Co.....Baltimore, Md.
McNary, I. R., Chief Engineer, U. S. N.....Navy Yard, Portsmouth, N. H.
Mechanics.....430 Walnut street, Philadelphia.
Mickley, J. P., Passed Assistant Engineer, U. S. N.....U. S. S. *Vesuvius.*
Milligan, R. W., Passed Assistant Engineer, U. S. N.....U. S. S. *Ranger.*
Miner, L. D., Assistant Engineer, U. S. N.....U. S. S. *Newark.*
Nagle, C. F., Passed Assistant Engineer, U. S. N.....U. S. S. *Michigan.*
Nauman, W. H., Passed Assistant Engineer, U. S. N.....U. S. S. *Petrel.*
Naval Institute.....Annapolis, Md.
Navy Department Library.....Navy Department, Washington, D. C.
Nones, H. B., Chief Engineer, U. S. N.....League Island Navy Yard, Philadelphia.
North East Coast Inst. of Engineers and Shipbuilders.....Newcastle-upon-Tyne, England.
Norton, H. P., Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Nystedt Thure, Marin Ingeniör-Statens Chefs Expedition.....Stockholm, Sweden.

Ogden, J. S., Passed Assistant Engineer, U. S. N.....U. S. S. *Despatch.*
Palmer, N. F., Presdt Quintard Iron Works.....12th street and Avenue D, New York.
Parks, W. M., Passed Assistant Engineer, U. S. N.....Manual Training School, Chicago.
Parsons, H. de B., Consulting Engineer.....35 Broadway, New York.
Patent Office Library (England).....25 Southampton Buildings, London, W. C.
Pemberton, J., Chief Engineer, U. S. N.....State College, Center Co., Pa.
Perry, J. H., Passed Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Pickrell, J. M., Assistant Engineer, U. S. N.....Richmond Locomotive Works, Richmond, Va.
Potts, Stacy, Passed Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Pray, Thos., Jr., Consulting Engineer.....P. O. Box 2728, Boston.
Pusey, C. W., General Manager Pusey & Jones Co.....Wilmington, Del.
Quintard, G. W., Quintard Iron Works.....12th street and Avenue D, New York.
Rae, C. W., Passed Assistant Engineer, U. S. N.....Care of Monroe & Co., 7 Rue Scribe, Paris.
Railroad and Engineering Journal.....P. O. Box 1515, New York.
Ransom, G. B., Passed Assistant Engineer, U. S. N.....University of Wisconsin, Madison, Wis.
Raynal, A. H., Supdg. Engineer Sam. L. Moore & Sons Co.....Elizabethport, N. J.
Rearick, P. A., Chief Engineer, U. S. N.....U. S. S. *Iroquois.*
Redgrave, DeW. C., Assistant Engineer, U. S. N.....Col. I. W. and D. D. Co., Baltimore, Md.
Reid, R. I., Passed Assistant Engineer, U. S. N.....U. S. S. *Newark.*
Rhoades, H. E., Assistant Engineer, U. S. N.....Mount Vernon, New York.
Roberts, E. E., Prop. Roberts' Boiler.....18 Cortlandt street, New York.
Robie, E. D., Chief Engineer, U. S. N.....Navy Yard, Norfolk, Va.
Robinson, L. W., Chief Engineer, U. S. N.....Member Board of Examiners, Philadelphia.
Roche, G. W., Chief Engineer, U. S. N.....Col. I. W. and D. D. Co., Baltimore, Md.
Roelker, C. R., Chief Engineer, U. S. N.....530 Twenty-first street, Washington, D. C.
Rommel, C. E., Assistant Engineer, U. S. N.....U. S. S. *Yorktown.*
Ross, H. S., Chief Engineer, U. S. N.....U. S. S. *Essex.*
Rowbotham, W., Passed Assistant Engineer, U. S. N.....U. S. S. *Baltimore.*
Rowland, T. F., Jr., Secretary and Treasurer Continental Iron Works...Station G, Brooklyn, N. Y.
Rozycki, S.....Wm. Cramp & Sons S. and E. B. Co., Philadelphia.
Salisbury, G. R., Assistant Engineer, U. S. N.....U. S. S. *Baltimore.*
Sampson, B. C., Assistant Engineer, U. S. N.....U. S. Naval Academy, Annapolis, Md.
Sanderson, R. P. C., Supt. of Motive Power N. & W. Ry.....Roanoke, Va.
Schell, F. J., Passed Assistant Engineer, U. S. N.....U. S. S. *Philadelphia.*
Schwartz, E. G., First Assistant Engineer, U. S. R. M.....U. S. R. S. *Manhattan.*
Scott, Irving M., General Manager Union Iron Works.....San Francisco, Cal.
Scribner, E. H., Assistant Engineer, U. S. N.....U. S. C. S. S. *Bache.*
See, Horace, Engineer and Naval Architect.....1 Broadway, New York.
Selden, W. C., Supdg. Engineer Clyde Line Steamers.....Pier 29, East River, New York.
Sims, Gardiner C., General Manager Armington and Sims Engine Co.....Providence, R. I.
Sims, W. Scott.....124 Water St., New York.
Sloane, J. D., Assistant Engineer, U. S. N.....Supervising Inspector's Office, St. Paul, Minn.
Smith, David, Chief Engineer, U. S. N.....
Member Steel Inspection Board, Navy Department, Washington, D. C.
Smith, J. A. B., Passed Assistant Engineer, U. S. N.....Col. Iron Works and D. D. Co., Balt., Md.
Smith, S. L., Supt. Roxbury Carpet Co.....37 Simmons street, Boston, Mass.
Smith, W. S., Chief Engineer, U. S. N.....U. S. S. *Marion*, Navy Yard, Mare Island, Cal.
Smith, W. Strother, Assistant Engineer, U. S. N.....Navy Yard, Norfolk, Va.
Smith, W. Stuart, Assistant Engineer, U. S. N.....Berkeley, Cal.
Spangler, H. W., Passed Assistant Engineer, U. S. N.....
Professor Dyn. Engineering, University of Pennsylvania, Phila.
Spiers, James, Mechanical Engineer...Fulton Iron Works, 213 Freemont street, San Francisco, Cal.
Steers, Henry, Naval Architect.....11th Ward Bank, New York.
Sternberg, H. S., Baron.....Military Attaché German Legation, Washington, D. C.
Stevenson, H. N., Passed Assistant Engineer, U. S. N....Delaware River Iron Works, Chester, Pa.
Strickland, G. D., Passed Assistant Engineer, U. S. N.....Standard Steel Works, Thurlow, Pa.
Taylor, D. W., Assistant Naval Constructor, U. S. N.....
Wm. Cramp & Sons S. and E. B. Co., Philadelphia, Pa.

Taylor, R. D., Passed Assistant Engineer, U. S. N.....3212 Hairnford street, Phila.
Taylor, Stevenson, V. P. and Gen'l Supt. of W. and A. Fletcher Co.....
12th and Hudson streets, Hoboken, N. J.
Tawresey, J. G., Assistant Naval Constructor, U. S. N.....Royal Naval College, Greenwich, Eng.
Theiss, Emil, Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Thomson, J. W., Chief Engineer, U. S. N.....Wm. Cramp & Sons S. and E. B. Co., Phila.
Tobin, J. A., Passed Assistant Engineer, U. S. N.....East Providence, R. I.
Tower, G. E., Chief Engineer, U. S. N.....U. S. S. *Kearsarge.*
Towne, N. P., Chief Engineer, U. S. N.....Bureau Steam Engineering, Navy Dept., Wash., D. C.
Trilley, Joseph, Chief Engineer, U. S. N.....Navy Yard, Portsmouth, N. H.
United Service Club.....Washington, D. C.
United Service.....1510 Chestnut St., Philadelphia.
Van Buren, J. D., Consulting Engineer.....Newburgh, N. Y.
Varney, W. H., Naval Constructor, U. S. N.....Col. I. W. and D. D. Co., Baltimore, Md.
Varney, W. W.....Col. I. W. and D. D. Co., Baltimore, Md.
Volz, W. E., Supdg. Engineer South Brooklyn Steam Engine Works.....Brooklyn, N. Y.
Voorhees, P. R., Counsellor-at-Law.....32 Nassau St., New York.
Warburton, E. T., Passed Assistant Engineer, U. S. N.....U. S. S. *San Francisco.*
Ward, Charles, Proprietor Ward Boiler.....Charleston, Kanawha County, W. Va.
Warren, B. H., Yale & Towne Manufacturing Co.....169 Falmouth street, Boston.
Warren, W. H., First Assistant Engineer, U. S. R. M., care of Collector of Customs, Detroit, Mich.
Warrington, J. N., Secretary Vulcan Iron Works.....Chicago, Ill.
Weaver, W. D., Assistant Engineer, U. S. N.....Hall Building, Troy, N. Y.
Webster, H., Passed Assistant Engineer, U. S. N.....Navy Department, Washington, D. C.
Weese, George.....97 Byrnes Roads, Partick, Glasgow, Scotland.
Wharton, B. B. H., Chief Engineer, U. S. N.....Navy Yard, Portsmouth, N. H.
Wheeler, F. M., Geo. F. Blake Mfg. Co. and Wheeler Condnsr. & Engng. Co., 93 Liberty st. N. Y.
Whittaker, E. J., Chief Engineer, U. S. N.....U. S. S. *Philadelphia.*
White, G. H., Chief Engineer, U. S. N.....U. S. S. *Richmond*, Newport, R. I.
White, W. W., Assistant Engineer, U. S. N.....U. S. C. S. S. *Blake.*
Williams, S. M.....13 Rutland Place, Govan, Glasgow, Scotland.
Williamson, J. D., Williamson Bros. E. and B. Works...York and Richmond streets, Philadelphia.
Williamson, Thom, Chief Engineer, U. S. N.....
Supt. State, War, and Navy Dept. Bldg., Navy Dept., Washington.
Willis, C. C., Assistant Engineer, U. S. N.....Orwood, Miss.
Willits, A. B., Passed Assistant Engineer, U. S. N.....43 Court street, Portsmouth, Va.
Wilmer, J. R., Passed Assistant Engineer, U. S. N.....U. S. Fish Commission Steamer *Albatross.*
Wilson, F. A., Chief Engineer, U. S. N.....U. S. S. *San Francisco.*
Winchell, W. P., Assistant Engineer, U. S. N.....U. S. S. *Fern.*
Windsor, W. A., Chief Engineer, U. S. N.....U. S. S. *Dolphin.*
Wood, B. F., Chief Engineer, U. S. N.....Continental Iron Works, Station G, Brooklyn.
Wood, J. L., Assistant Engineer, U. S. N.....Union Iron Works, San Francisco, Cal.
Woods, A. T., Professor Mechanical Engineering.....University of Illinois, Champaign, Ill.
Woodward, J. J., Naval Constructor, U. S. N.....Navy Department, Washington, D. C.
Worthington, C. C., Hydraulic Engineer.....H. R. Worthington, 145 Broadway, New York.
Worthington, W. F., Passed Assistant Engineer, U. S. N.....U. S. S. *Atlanta.*
Yarrow, A. F., Naval Architect and Mechanical Engineer.....Isle of Dogs, Poplar, London.
Zeitschrift des Vereines Deutscher Ingenieure.....131 Potsdamer street, Berlin.
Zeller, Theo., Chief Engineer, U. S. N.....15 West 12th street, New York.
Zinnell, G. F., Prof.....816 North 26th street, Philadelphia.

Members are requested to notify the Secretary-Treasurer of any mistakes made.

Discussion is invited on the papers on "Preservation of Marine Boilers" and on "Engine Room Signals." Remarks received already are held over for the August number of the JOURNAL.